TRANSACTIONS

THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

VOLUME 56

Fifty-Sixth Annual Meeting
Dallas, Tex.
January 23-26, 1950

Semi-Annual Meeting
Muskoka Lakes, Ont., Canada
June 19-21, 1950

PUBLISHED BY THE SOCIETY AT THE OFFICE OF THE SECRETARY
51 Madison Avenue
NEW YORK 10, N. Y.
Printed in U.S.A.

Printed and bound by
The Horn-Shafer Co.
Baltimore, Md.
U.S.A.

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Treasurer.....C. E. Wynn
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Treasurer.....D. L. Bergman

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* Filled unexpired term.

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Organized 1948

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Treasurer.....J. M. Black
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Organized 1935

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Organized 1922

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Organized 1939

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Secretary.....W. B. Pride
Treasurer.....R. R. Kirkwood
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Organized 1916

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2nd Vice-President.....M. E. Barnard
Secretary.....L. M. Church
Treasurer.....C. F. Dietz
Board of Governors: M. E. Barnard, F. H. Buzzard, L. M. Church, C. F. Dietz, J. W. McElgin, W. S. Scott, E. K. Wagner

Pittsburgh

Organized 1919

Headquarters, Pittsburgh

President.....A. F. Metzger
Vice-President.....W. D. Simpson
Secretary.....E. H. Riesmeyer, Jr.
Treasurer.....B. B. Reilly
Board of Governors: H. A. Biber, H. J. Kirkendall, B. R. Small

Rocky Mountain

Organized 1944

Headquarters, Denver, Colo.

President.....R. W. Peterson
Vice-President.....B. H. Spurlock, Jr.
Secretary.....P. C. Von Rosenberg
Treasurer.....P. W. Young
Board of Governors: A. W. Cooper, Fred Janssen, E. J. McEahern

St. Louis

Organized 1918

Headquarters, St. Louis

President.....J. S. Rosebrough
1st Vice-President.....C. H. Burnap
2nd Vice-President.....H. C. Sharp
Secretary.....L. L. Hamig
Treasurer.....G. H. Bemarkt
Board of Governors: J. F. Naylor, W. A. Russell, Louis Steckhan, H. F. Wilson

Shreveport

Organized 1948

Headquarters, Shreveport

President.....B. E. Segall, Jr.
Vice-President.....S. W. Beaty
Secretary.....R. M. Hood
Treasurer.....W. S. Evans
Board of Governors: S. W. Beaty, W. S. Evans, W. E. FitzGerald, R. M. Hood, A. H. Otto, P. O. Rottmann, B. E. Segall, Jr.

South Texas

Organized 1938

Headquarters, Houston

President.....H. W. Broadwell
Vice-President.....G. J. Collins
Secretary.....C. L. Fleming
Treasurer.....E. G. Floeter, Jr.
Board of Governors: J. W. Holland, E. G. Floeter, Jr., R. J. Salinger

Southern California

Organized 1930

Headquarters, Los Angeles

President....L. J. Helms
Vice-President....M. C. Greiner
Secretary....J. S. Earhart
Treasurer....L. B. Davenport
Board of Governors: J. L. Blake, R. S. Farr, A. J. Hess, Harry Kunkle, R. A. Lowe

Southwest Texas

Organized 1946

Headquarters, San Antonio

President....L. S. Pawkett
Vice-President....I. W. Wilke
Secretary-Treasurer....W. E. Long
Board of Governors: G. R. Rhine, A. J. Rummel, F. C. Benham

Utah

Organized 1944

Headquarters, Salt Lake City

President....R. H. East
Vice-President....A. R. Curtis
Secretary-Treasurer....D. R. Wilde
Board of Governors: C. E. Ferguson, E. V. Gritton, D. B. Holford, E. J. Watts

Virginia

Organized 1946

Headquarters, Norfolk

President....W. P. Robinson
Vice-President....J. F. Boyenton
Secretary....D. E. Phillips
Treasurer....J. E. Harding
Board of Governors: E. D. Duval, J. J. Shanahan, J. E. White

Washington, D.C.

Organized 1935

Headquarters, Washington, D. C.

President....F. M. Thuney
Vice-President....R. K. Thulman
Secretary....I. M. Bortman
Treasurer....P. H. Loughran, Jr.
Board of Governors: P. R. Achenbach, J. G. Muirheid, W. C. Reamy, Jr.

Western Michigan

Organized 1931

Headquarters, Grand Rapids

President....K. E. Robinson
Vice-President....V. H. Hill
Secretary....F. W. Brundage
Treasurer....W. C. DeRoo
Board of Governors: J. L. Alexander, L. A. Calcaterra, H. R. Limbacher

Western New York

Organized 1919

Headquarters, Buffalo

President....J. H. Bryce
1st Vice-President....T. F. Killeen
2nd Vice-President....J. M. Quackenbush
Secretary....C. W. Stone
Treasurer....B. C. Candee
Board of Governors: M. C. Beman, Joseph Davis, Roswell Farnham, S. M. Quackenbush, F. J. Weber

Wisconsin

Organized 1922

Headquarters, Milwaukee

President....F. J. Nunlist
Vice-President....H. F. Brinen
Secretary....N. E. Hill
Treasurer....J. A. Lofte
Board of Governors: B. M. Kluge, O. A. Trostel, J. R. Vernon

STUDENT BRANCHES

College of City of New York

Organized 1949

Headquarters, New York

President....S. Oestreicher, S. Weinberg
Vice-President....S. Weinberg, D. Weinberg
Corresponding Secretary..D. Weinberg, B. Karp
Recording Secretary..M. Goldman, N. Burtman
Treasurer....S. Gardner, D. Weinberg

Louisiana Polytechnic Institute

Organized 1949

Headquarters, Ruston

President....C. W. Gorton, H. A. Parker
Vice-President....G. F. Lomax, Jr., J. A. Adams
Secretary....A. L. Jones, E. L. Lentz
Treasurer....D. C. Cummings, Jr., B. C. Glenn

North Carolina State College

Organized 1948

Headquarters, Raleigh

President....L. R. Wensil, Jr.
Vice-President....Sydney Narvey
Secretary....V. V. Crews, F. T. Lathers*
Treasurer....E. R. Malin
Reporter....S. W. Thomason

* Filled unexpired term.

Oregon State College

Organized 1949

Headquarters, Corvallis

President....R. J. Kemper
Vice-President....F. E. Schroeder
Secretary-Treasurer....J. M. Templer

Student Branch Officers—1950 *(Continued)*

Purdue University

Organized 1948

Headquarters, W. Lafayette, Ind.

Chairman..........D. M. Williams, J. J. Harrell
Vice-Chairman..........R. L. Ford
Secretary..........J. D. Tuttle, June Cox
Reporter..........G. R. Markstrom
Coordinator..........J. J. Harrell

Texas A. & M.

Organized 1946

Headquarters, College Station

President..........J. O. Kadel
Vice-President..........W. E. Weatherby
Secretary..........W. C. Haggard
Treasurer..........Charles Luedtke, Jr.

University of Detroit

Organized 1949

Headquarters, Detroit

President..........L. M. Blanchette
Vice-President..........A. F. Lewis
Secretary-Treasurer..........S. H. Kosinski
Reporter..........R. E. Seaton

University of Kansas

Organized 1949

Headquarters, Lawrence

President..........J. L. Williams
Vice-President..........George Saller
Secretary-Treasurer..........Robert Umholtz

University of Texas

Organized 1949

Headquarters, Austin

President..........L. T. Davis, L. E. Scott
Vice-President..........L. E. Scott, R. M. Tittle
Secretary..........A. G. Allen, E. F. Luker
Treasurer..........F. T. Saade, J. C. Tittle

1377

TRANSACTIONS

of

The American Society of Heating and Ventilating Engineers

No. 1377

FIFTY-SIXTH ANNUAL MEETING, 1950

Dallas, Texas

The 56th Annual Meeting of The American Society of Heating and Ventilating Engineers was held in Dallas, Tex., January 22-26, 1950, and the attendance of 2157 members and guests, which was the second largest in Society history, exceeded the expectations of all except the Committee on Arrangements of North Texas Chapter, which, with true Texas optimism, had been certain that this would be the Society's largest meeting. The official registration figures were: members 935, guests, 760, ladies 462.

Two hotels, the Baker and the Adolphus, were used as the meeting headquarters, and both hotels were used for technical sessions as well as other events. Five technical sessions were held.

Entertainment features included: a welcome luncheon at the Adolphus Hotel on January 23, at which Stanley Foran, was the speaker and P. N. Vinther was the toastmaster; a Chuck Wagon Dinner in the Foods Building at the State Fair Grounds, at 7:00 p.m. on January 23; a luncheon at the Adolphus Hotel on January 24, with K. C. Richmond, New York, N. Y. editor of *Coal-Heat,* speaking on *The Human Equation—A Public Relations Problem in Heating and Air Conditioning;* the Annual Banquet in the ballroom at the Baker Hotel, with C. Rollins Gardner, general chairman of the Committee on Arrangements, as toastmaster, and Dr. Umphrey Lee, president of Southern Methodist University, as the principal speaker. Dr. Lee's subject was *The Fearful Era.*

Events of special significance at the Annual Banquet were the awarding of the F. Paul Anderson Medal to Dr. C.-E. A. Winslow, the presentation being made by Pres. Alfred E. Stacey, Jr.; the presentation of the Past-President's Emblem to Mr. Stacey by Walter L. Fleisher, past-president of the Society; and presentation of the Memory Book to Past-President G. L. Tuve, made by Mr. Fleisher.

The Southwest Air Conditioning Exposition at the State Fair Grounds, was opened at 2:00 p.m. on January 23, by R. L. Thornton, president of the State Fair of Texas, with participation in the ceremony by Pres. A. E. Stacey, Jr., Syracuse, N. Y., and other Society Officers.

FIRST SESSION, MONDAY, JANUARY 23RD, 9:30 A.M.

President Stacey called the meeting to order in the Ballroom of the Baker Hotel, and introduced P. N. Vinther, chairman of the sessions committee, who welcomed the members and guests.

President Stacey acknowledged the cordial welcome and called for reports of the Officers and Council. President Stacey gave the President's report.

REPORT OF PRESIDENT

The year's activities of the Society are to be fully reported by the chairmen of the several committees and so it seems to me to be totally unnecessary to anticipate these gentlemen. However, I wish to call your attention to a matter which is of outstanding importance. That is the rapid growth of the Society over the last 15 years. We have grown since 1935 from a membership of less than 2000 to our present gratifying membership of 7700. This is a gain of approximately 400 percent. Because of this extraordinary growth, our expenditures for the membership have increased at a correspondingly great rate, and the budget for 1950 is over $450,000. This is a large amount of money to be spent wisely and to the best interests of the Society. It is a heavy responsibility of your Council to see that this is done. From time to time, it has been necessary to meet the new situations resulting from this growth by Council action. Your Constitution and By-Laws Committee have based the revision of the By-Laws, of which you have all received notice, upon these Council actions and those suggestions of committees which, of course, have met the approval of the Council.

One of a number of my pleasures and duties as President has been to visit many chapters of the Society. Nothing was left undone by these Chapters to make me welcome—I had the pleasure of making new friends and renewing old acquaintances. Best of all was the enthusiasm and great interest in the affairs of the Society which I found displayed throughout the country. I had not realized to what extent the Society had grown.

Every president of any organization feels that his accomplishments would be very feeble were it not for the cooperation and hard work of a very great many people. It is impossible for me to express my gratitude to the Council Members, Chairmen of Committees, Presidents of Chapters, the Staff Members of the Executive Office and the Laboratory, and a host of faithful members who have combined to bring about any success which this year has held. It has been a privilege to work with you all as your president.

Respectfully submitted,

ALFRED E. STACEY, JR.

REPORT OF COUNCIL

The new Council held its first meeting in Chicago, Ill., January 27, 1949, with Pres. Alfred E. Stacey, Jr., presiding. He announced the personnel of the Council, General and Special Committees, the appointments of which were confirmed.

Appointment of the secretary, assistant to president, certified public accountant and legal counsel were made and the depositories for Society funds in New York, Cleveland, and Toronto were approved. New membership application and reference forms were approved and a procedure for expediting the election of student members was adopted.

A special Committee on Membership Grades reported its recommendations which were referred to the Constitution and By-Laws Committee and a Special Committee on Clarification of Committee Functions outlining the scope of activities for the Chapter Relations Committee and the Chapter Delegates Committee. As required

by the By-Laws, five candidates were nominated for a three-year term on the Committee on Research and four representatives and one alternate were selected for service on the Nominating Committee.

The time and place of the Semi-Annual Meeting, 1950, and the Annual Meeting, 1951, were announced respectively, at the Royal Muskoka Hotel, Ontario, June 19-21, with the Ontario Chapter as host and at Philadelphia, Pa., January 22-25, with the Philadelphia Chapter as host. Approval was given to the Programs of the Minneapolis and Dallas Meetings as submitted by the Program and Papers Committee. An exhibit by the Society featuring its general activities and research work was approved for the Dallas exhibition. Council approved the plan of having expositions on a biennial basis after 1951.

A budget for the fiscal period November 1, 1949, to October 31, 1950, was adopted with estimated receipts of $427,790.00 and disbursements of $454,250.00. A Research Reserve Fund was created upon recommendation of the Finance Committee. Re-classification of the Society's status under the Internal Revenue Code is pending. The Council accepted with regret the resignation of Clyde A. McKeeman, as assistant to the president, and the resignation of Prof. M. C. Giannini, as a member of the Committee on Research. P. B. Gordon was appointed to serve in Professor Giannini's place. A special meeting of the Society was held in New York, November 21, at which time the amended Charter* was approved and it was voted to submit the new By-Laws to the Society for vote at the 56th Annual Meeting. The By-Laws (see Chapter 1378) were prepared by the Constitution and By-Laws Committee, submitted to Council in October, and a number of amendments by Council were incorporated in the final draft.

The award of the F. Paul Anderson Medal was approved and Dr. C.-E. A. Winslow was nominated for the award. Re-appointment of the Director of Research was voted.

Charters for two chapters and seven student branches were authorized.

During the year, Council voted on the admission of members recommended by the Admission and Advancement Committee and authorized Life Membership for eligible candidates. Action was taken on members who submitted resignations and cancellation of memberships for non-payment of dues was voted.

Respectfully submitted,

THE COUNCIL

REPORT OF THE SECRETARY

The year 1949 has been one of exceptional activity at the headquarters office with many extra duties required in connection with the growth of membership, which is now at an all time high, the revision of the Charter and streamlining the By-Laws, more active committee work relating to chapters, Guide expansion, meetings, public relations and other administrative duties assigned by Council.

The financial statement reflects the expansion of Society activities, the membership report shows an increase in membership, THE GUIDE enjoyed a larger income but a small reduction in distribution.

The accounting system was completely overhauled and adapted to a machine bookkeeping set-up which became effective in November. New record cards for dues, address forms, a new type of duesbill were developed and installed. The result will cut down some of the operations, give quicker totals and provide better financial records.

Contact has been maintained with chapter officers and whenever practical visits have been made at regular meetings or with the officers. During the year the chapters

* Charter (A.S.H.V.E. TRANSACTIONS, Vol. 55, 1949, p. 489).

responsible for the arrangements at the Semi-Annual and Annual Meetings at Minneapolis and Dallas respectively have worked closely with the Secretary's office and are to be commended for their work in planning and carrying out the great number of details required in making a successful meeting.

All of the Council Committees, Special Committees and General Committees have carried on their functions as required by the operating rules of the Society and it is a distinct pleasure to acknowledge their interest and cooperation.

My staff of associates at the headquarters office deserve an extra tribute for their loyal interest in the Society and their effective work, often under severe handicaps.

Respectfully submitted,

A. V. HUTCHINSON,

Secretary

Dean Seeley, second vice president of the Society, gave a brief review of the status of Society finances, indicating that the Society was in excellent condition. He stated that the auditor's report would be published.

Accountant's Report

FRANK G. TUSA & CO.

CERTIFIED PUBLIC ACCOUNTANTS

37 Wall St., New York 5, N. Y.

THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
51 Madison Ave.,
New York, N. Y.

Gentlemen:

Pursuant to your request, we examined the books of account and records of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS—New York, N. Y. and the related funds for the fiscal year ended October 31, 1949 and submit herewith our report.

The audit covered a verification of the Assets and Liabilities as of the close of business October 31, 1949. Also for the fiscal year then ended, the recorded cash receipts were traced into the depositories; the cancelled bank checks were inspected and compared with the record of cash disbursements; the disbursements were supported by payment vouchers and the dues income from members and the interest income from investments was accounted for.

A balance sheet reflecting the financial condition of the Society as of the close of business October 31, 1949 is submitted herewith and your attention is directed to the following comments thereon:

CASH

The cash on deposit was verified by direct communication with commercial and savings banks and the balances reported to us were reconciled with those reflected by the books of the Society. A schedule of cash is included as a part of this report.

Checks representing the cash on hand for deposit were inspected by us and the petty cash was counted.

MARKETABLE SECURITIES

The securities shown on the subjoined schedule were verified by direct communication with the Bankers Trust Co., where same are deposited for safekeeping. This asset has been included in the balance sheet at the cost of acquisition plus the accumulated and accrued interest earned thereon.

ACCOUNTS RECEIVABLE

Trial balance taken of the membership dues receivable and sundry debtors as of the close of business October 31, 1949 were classified and aged as follows:

DUES RECEIVABLE

MEMBERSHIP	Amount
Members	$ 4,100.92
Associates	8,967.84
Juniors	668.00
Students	143.00
TOTAL	$13,879.76
AGING	
1949	$13,379.56
1948	438.20
1947	62.00
TOTAL	$13,879.76

SUNDRY DEBTORS

Unpaid charges made during October, 1949	$5,413.30
Unpaid charges made during September, 1949	1,728.26
Unpaid charges made prior	1,197.93
TOTAL	$8,339.49

After writing off all receivables known to be uncollectible, it is our opinion that the reserves for dues and accounts receivable doubtful of collection reflected in the accompanying balance sheet are ample to cover collection losses that may be incurred.

INVENTORIES

The emblems and TRANSACTIONS on hand on October 31, 1949 were counted by us. THE GUIDE paper was verified by direct communication with the printers. All inventories were priced and computed by us.

A schedule of TRANSACTIONS inventoried follows:

Volume	Year	Quantity	Price	Amount
1-48	1895-1942	3205	$0.40	$1,282.00
49	1943	231	1.39	321.09
50	1944	189	1.82	343.98
51	1945	348	1.91	664.68
52	1946	417	1.67	696.39
53	1947	327	1.77	578.79
TOTALS		4717		$3,886.93

DEPOSIT RECEIVABLE

The deposit placed with the United Air Lines in the sum of $425.00 was verified by direct communication.

ADVANCES

We have classified under exchanges advances made for the accommodation of Council members reimbursed subsequently to the date of audit.

As at October 31, 1949 the participating employees to the retirement plan were indebted in the sum of $1,433.46 for contributions advanced by the Society to the pension fund. This sum is to be reimbursed to the Society by periodical deductions from wages of employees covered by the plan.

DEFERRED CHARGES

One-sixth of all subscriptions paid during the current fiscal period to HPAC were deferred as a future expense since the subscriptions are paid on a calendar year basis and the fiscal year of the Society ended on October 31, 1949.

TAXES

The sum of $465.10 represents federal income taxes withheld from salaries paid to employees during the month of October, 1949. The Society may be subject to possible social security tax assessments pending decision from the U.S. Treasury Department.

DUE RESEARCH

The balance due the Research Laboratory as at October 31, 1949 is made up as follows:

40 Percent of Dues Receivable from Members and Associates		$5,227.50
Less: 40 Percent of Dues Prepaid by Members and Associates	$419.91	
Amount advanced to George Hastings for fund raising expenses on behalf of Research	40.54	460.45
BALANCE		$4,767.05

ACCRUED ACCOUNTS

The salaries and commissions accrued as at October 31, 1949 have been computed by us in accordance with the Finance Committee report of April 8, 1948. Other accruals cover various expenses applicable to the current fiscal year.

DEFERRED INCOME

The prepaid dues and initiation fees by members and candidates for membership have been deferred to future operations. The membership classification of the dues prepaid by elected members follows:

MEMBERSHIP	Amount
Members	$ 505.18
Associates	544.59
Juniors	344.67
Students	62.00
TOTAL	$1,456.44

RESERVE FOR TRANSACTIONS

We have reserved the sum of $24,500.00 to cover the publication of TRANSACTIONS Volumes 54 and 55 that are scheduled for publication in 1950.

FUNDS

An analysis of the following fund accounts covering the changes that occured therein during the fiscal year ended October 31, 1949 is attached hereto:

General Fund
Reserve Fund
F. Paul Anderson Fund
Property Fund
Mortgage Reduction Fund

In accordance with the resolution adopted by the Council at its meeting of January 23, 1949 depreciation was not provided on the buildings for the current fiscal year.

Respectfully submitted,

FRANK G. TUSA & CO.
Certified Public Accountants

BALANCE SHEET

THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS—New York, N. Y.

October 31, 1949

ASSETS

GENERAL FUND				
CASH				
On Deposit		$52,927.51		
On Hand		100.00	$53,027.51	
SECURITIES (AT COST)				
Securities (Market Value $47,005.59)		43,610.75		
Add: Accumulated Interest	$ 4,149.84			
Add: Accrued Interest	182.90	4,332.74	47,943.49	
ACCOUNTS RECEIVABLE				
Membership Dues	13,879.76			
Less: Reserve for Doubtful	6,000.00	7,879.76		
Advertisers and Sundry Debtors	8,339.49			
Less: Reserve for Doubtful	1,000.00	7,339.49	15,219.25	
INVENTORIES				
TRANSACTIONS		3,886.93		
Emblems		448.75		
GUIDE Paper		335.00	4,670.68	
DEPOSITS RECEIVABLE			425.00	
EXCHANGES			1,359.16	
ADVANCES TO PENSION FUND FOR EMPLOYEES			1,433.46	
DEFERRED CHARGES				
Prepaid HPAC Subscriptions		2,432.46		
Prepaid Advertising Sales Promotion		900.00		
Unexpired Insurance Premiums		248.70	3,581.16	$127,659.71
PROPERTY FUND				
Land and Buildings		84,920.96		
Less: Reserve for Depreciation		3,908.07	81,012.89	
Furniture and Fixtures		7,389.17		
Less: Reserve for Depreciation		1,556.17	5,833.00	
Library			300.00	87,145.89
RESERVE FUND				
Cash on Deposit			3,563.80	
Securities at Cost (Market Value $46,436.20)		40,145.00		
Add: Accumulated Interest		6,291.20	46,436.20	50,000.00
F. PAUL ANDERSON FUND				
Cash on Deposit		86.08		
Cash on Hand for Deposit		25.00	111.08	
Securities at Cost (Market Value $947.00)		1,000.00		
Add: Accrued Interest		12.50	1,012.50	1,123.58
				$265,929.18

LIABILITIES AND FUNDS

LIABILITIES				
FEDERAL WITHHOLDING TAX			$ 465.10	
DUE RESEARCH			4,767.05	
ACCRUED ACCOUNTS				
Salaries and Commissions		$ 7,895.10		
Expenses		4,313.37	12,208.47	
DEFERRED INCOME				
Prepaid Membership Dues:				
Elected Members	$ 1,456.44			
Candidates	1,092.75	2,549.19		
Prepaid Initiation Fees		1,679.00		
Past President's Award		100.00	4,328.19	
RESERVE FOR TRANSACTIONS				
TRANSACTIONS—Volume 53		1,651.85		
TRANSACTIONS—Volume 54		12,000.00		
TRANSACTIONS—Volume 55		12,500.00	26,151.85	
TOTAL LIABILITIES				$ 47,920.66
FUNDS				
Property Fund			87,145.89	
Reserve Fund			50,000.00	
F. Paul Anderson Fund			1,123.58	
General Fund			79,739.05	
NET WORTH OF SOCIETY				218,008.52
				$265,929.18

NOTE: This balance sheet is subject to possible assessments for Social Security taxes pending decision by the Treasury Department.

BUDGET COMPARISON—CASH RECEIPTS

THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS—New York, N. Y.

For the Fiscal Year Ended October 31, 1949

DUES	Actual	Budget Provision	Increases	Decreases
CURRENT MEMBERS				
6A—Members	$ 38,308.99	$ 39,000.00	$	$ 691.01
6B—Associates	34,743.13	36,000.00		1,256.87
6C—Juniors	4,695.06	5,000.00		304.94
6D—Students	961.00	900.00	61.00	
	78,708.18	80,900.00	61.00	2,252.82
NEW MEMBERS				
104—Members	1,223.30	1,875.00		651.70
105—Associates	2,249.85	1,500.00	749.85	
106—Juniors	1,920.00	500.00	1,420.00	
107—Students	1,148.00	150.00	998.00	
	6,541.15	4,025.00	3,167.85	651.70

PRIOR YEARS DUES				
6E—Members	1,024.00	1,200.00		176.00
6F—Associates	1,443.04	1,500.00		56.96
6G—Juniors	182.50	400.00		217.50
6H—Students	18.00	60.00		42.00
	2,667.54	3,160.00		492.46
TOTAL DUES	87,916.87	88,085.00	3,228.85	3,396.98
108—INITIATION FEES	5,486.00	5,500.00		14.00
DEFERRED INCOME				
Prepaid Membership Dues	1,456.44	1,000.00	456.44	
Prepaid Dues by Candidates	1,092.75	200.00	892.75	
108A—Prepaid Initiation Fees	1,679.00	1,000.00	679.00	
	4,228.19	2,200.00	2,028.19	
PUBLICATIONS				
115—JOURNAL Contract	24,333.32	23,500.00	833.32	
300—GUIDE Advertising	60,241.01	56,000.00	4,241.01	
302—GUIDE	59,024.99	50,000.00	9,024.99	
117—TRANSACTIONS	861.93	1,500.00		638.07
118—Books, Reprints and Codes	2,104.39	700.00	1,404.39	
	146,565.64	131,700.00	15,503.71	638.07
INVESTMENT INCOME				
125—Interest—Savings Bank	332.45	100.00	232.45	
126—Interest—Securities	521.01	400.00	121.01	
	853.46	500.00	353.46	
OTHER INCOME				
109—Sale of Emblems	978.16	1,500.00		521.84
110—Sale of Certificate Frames	526.00	500.00	26.00	
	1,504.16	2,000.00	26.00	521.84
RESEARCH LABORATORY RECEIPTS				
40 Percent of Members and Associates Dues	53,718.75	54,050.00		331.25
Contributions—Allocated	56,477.50	60,000.00		3,522.50
Contributions—General	8,635.00	10,000.00		1,365.00
Navy Research	21,042.49	59,375.00		38,332.51
Exposition—1949	26,718.10	24,500.00	2,218.10	
Exposition—1948	50.00	—0—	50.00	
Interest	6.76	—0—	6.76	
Bulletins	92.50	—0—	92.50	
	166,741.10	207,925.00	2,367.36	43,551.26
TOTAL RECEIPTS	$413,295.42	$437,910.00	$23,507.57	$48,122.15

BUDGET COMPARISON—CASH DISBURSEMENTS

THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS—New York, N. Y.

October 31, 1949

COMMITTEES AND CHAPTERS	Actual	Budget Provision	Increases	Decreases
150—President's Fund	$ 952.40	$ 3,000.00	$	$ 2,047.60
151—Council Travel	2,680.80	7,200.00		4,519.20
160—Executive Committee	122.14	100.00	22.14	
161—Finance Committee	95.99	100.00		4.01
162—Membership Committee		1,000.00		1,000.00
164—Standards Committee		150.00		150.00
170—Admissions and Advancement Committee	635.55	1,000.00		364.45
171—Constitution and By-Laws	143.37	1,000.00		856.63
172—Nominating Committee	1,292.54	1,500.00		207.46
173C—Chapter Relations Committee	511.55	500.00	11.55	
173A—Chapter Speakers	1,584.54	3,000.00		1,415.46
173B—Chapter Delegates	3,227.97	5,000.00		1,772.03
201—A.S.A. Membership	100.00	100.00		
204—Membership Certificates	1,654.16	750.00	904.16	
204A—Framed Certificates	335.00	250.00	85.00	
205—Emblems	684.65	1,500.00		815.35
206—Medals and Awards	71.40	250.00		178.60
	14,092.06	26,400.00	1,022.85	13,330.79
MEETINGS				
163—Meetings	3,536.50	4,500.00		963.50
327—Chapter Meeting Allowance	600.00	2,000.00		1,400.00
	4,136.50	6,500.00	—0—	2,363.50
PUBLICATIONS				
200—JOURNAL Subscriptions	14,808.99	14,000.00	808.99	
202—TRANSACTIONS—Volume 53	10,490.37	12,000.00		1,509.63
TRANSACTIONS—Volume 54		12,000.00		12,000.00
203—Membership Roll	5,376.07	2,500.00	2,876.07	
305—326—GUIDE Publication and Distribution	68,782.97	70,000.00		1,217.03
118A—Books, Reprints, Codes	1,743.98		1,743.98	
	101,202.38	110,500.00	5,429.04	14,726.66
HEADQUARTERS				
210—Salaries	59,282.86	64,000.00		4,717.14
210A—Contingent Fund		2,000.00		2,000.00
211—Public Relations Expense	4,544.40	5,000.00		455.60
212—Travel—Secretary and Staff	3,123.15	3,000.00	123.15	
213—Rent and Light	5,329.56	5,600.00		270.44
214—Telephone	1,548.06	1,200.00	348.06	
215—Telegraph	399.94	500.00		100.06
216—Postage	4,446.52	4,000.00	446.52	
217—Printing and Stationery	3,451.18	3,500.00		48.82
219—Addressing and Address Changes	365.47	300.00	65.47	
220—Professional Services	3,410.47	3,500.00		89.53

Headquarters (*continued*)

221—Bank Charges	57.76	75.00		17.24
222—Insurance	457.57	1,000.00		542.43
11—Furniture and Fixtures	4,233.95	4,500.00		266.05
223—General Office Expenses	1,205.89	1,500.00		294.11
224—Pension	1,434.76	1,500.00		65.24
	93,291.54	101,175.00	983.20	8,866.66
Research				
Research Committee Expenses	1,926.25	3,000.00		1,073.75
Staff Salaries	77,699.77	95,400.00		17,700.23
Laboratory Expenses	24,209.42	45,000.00		20,790.58
Laboratory Operations and Maintenance	11,319.54	12,000.00		680.46
Cooperative Research	14,041.00	20,600.00		6,559.00
Navy Research	16,827.90	46,500.00		29,672.10
	146,023.88	222,500.00	—0—	76,476.12
Total Disbursements	$358,746.36	$467,075.00	$7,435.09	$115,763.73

Unbudgeted Disbursements

Investments	$ 9,171.25
Final Mortgage Payment	7,000.00
Loss on Canadian Exchange	1,537.71
Accounts Receivable and Exchanges	959.89
Federal Withholding Taxes	61.50
Hospitalization Advances—Research	82.60
	$ 18,812.95

BALANCE SHEET

The American Society of Heating and Ventilating Engineers

Research Fund—New York, N. Y.

October 31, 1949

ASSETS

Research Fund			
Cash			
On Deposit			
Treasurer's Account—Bankers Trust Co.	$ 16,231.70		
Director's Account—Cleveland Trust Co.	5,237.03		
Travel Account—Cleveland Trust Co.	500.00		
Thrift Account—Bank for Savings	458.27	$ 22,427.00	
In Transit		2,946.31	
On Hand for Deposit		11,000.16	
On Hand		250.00	$ 36,623.47
Accounts Receivable			
A.S.H.V.E. (40 Percent of Dues when Collected)		4,767.05	
U.S. Navy Department		5,112.01	
Advances to Employees:			
Retirement Plan	976.97		
Hospitalization	88.20	1,065.17	10,944.23

DEPOSIT RECEIVABLE			
Drums			12.00
PERMANENT			
Laboratory Equipment, Furniture and Fixtures	34,988.06		
Less: Reserve for Depreciation	14,436.63	20,551.43	
Tools		300.00	20,851.43
DEFERRED CHARGES			
Unexpired Insurance Premiums			1,506.14
TOTAL RESEARCH FUND			69,937.27
RESERVE FUND			
CASH ON DEPOSIT			
Bankers Trust Co.		4,000.00	
Central Bank for Savings		5,300.00	
Dime Savings Bank		5,518.10	
East River Savings Bank		5,300.00	
Emigrant Industrial Savings Bank		5,300.00	
Seaman's Bank for Savings		5,300.00	
TOTAL RESERVE FUND			30,718.10
ENDOWMENT FUND			
CASH ON DEPOSIT			
Bank for Savings			659.17
			$101,314.54

LIABILITIES AND FUNDS

LIABILITIES			
ACCOUNTS PAYABLE		$ 118.75	
FEDERAL WITHHOLDING TAX		.50	
DEFERRED INCOME			
Sound Energy Studies	$ 574.39		
Heat Losses Due to Infiltration	188.75		
Panel Heating Studies	8,585.00		
Odor Studies	3,550.00		
Heat Flow Through Glass	5,000.00		
Miscellaneous	500.00	18,398.14	
EXCHANGES		8.39	$ 18,525.78
FUNDS			
Property Fund		20,851.43	
Reserve Fund		30,718.10	
Endowment Fund		659.17	
Research Fund		30,560.00	82,788.76
			$101,314.54

NOTE "A"—This balance sheet may be subject to possible assessments for Social Security taxes pending final decision by the Treasury Department.

BUDGET COMPARISON—CASH RECEIPTS AND DISBURSEMENTS

THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

RESEARCH FUND—New York, N. Y.

For the Fiscal Year Ended October 31, 1949

CASH RECEIPTS	Actual	Budget Provision	Increases	Decreases
40 Percent Dues—Members and Associates	$ 53,718.75	$ 54,050.00	$	$ 331.25
Heating and Ventilating Exposition—1949	26,718.10	24,500.00	2,218.10	
Heating and Ventilating Exposition—1948	50.00		50.00	
Allocated Contributions—1949	56,477.50	60,000.00		3,522.50
General Contributions	8,635.00	10,000.00		1,365.00
Office of Naval Research	21,042.49	59,375.00		38,332.51
Interest	6.76		6.76	
Sale of Bulletins	92.50		92.50	
TOTALS	$166,741.10	$207,925.00	$ 2,367.36	$ 43,551.26
ALLOCATED CONTRIBUTIONS—1949				
Heat Transfer Through Glass	$ 15,330.00	$ 22,000.00	$	$ 6,670.00
Heat Transfer—General	450.00		450.00	
Physiological Studies	4,171.00	6,000.00		1,829.00
Air Cleaning	6,250.00	6,000.00	250.00	
Air Flow—General	3,700.00	2,000.00	1,700.00	
Panel Heating and Cooling	18,526.50	20,000.00		1,473.50
Odors	3,550.00		3,550.00	
Miscellaneous	4,500.00	4,000.00	500.00	
TOTALS	$ 56,477.50	$ 60,000.00	$ 6,450.00	$ 9,972.50
CASH DISBURSEMENTS				
COMMITTEE EXPENSES				
100—Travel and Meeting Expenses, Chairman, Research Executive Committee and Technical Advisory Committees	$ 1,598.25	$ 2,500.00		901.75
101—Chairman's Office Expense	328.00	500.00		172.00
	1,926.25	3,000.00	—0—	1,073.75
LABORATORY SALARIES				
150—Administrative and Technical Staff	65,145.46	75,600.00		10,454.54
151—Clerical Staff	10,994.59	10,800.00	194.59	
152—Pension Contributions	933.46	1,500.00		566.54
153—Part Time Staff and Contingencies	626.26	7,500.00		6,873.74
	77,699.77	95,400.00	194.59	17,894.82
LABORATORY EXPENSES				
160—Staff Travel	4,523.17	4,800.00		276.83
161—Postage	500.54	1,000.00		499.46
162—Telephone and Telegraph	1,714.82	1,800.00		85.18
163—Office Supplies	1,234.66	1,200.00	34.66	
164—Mimeographing and Printing	1,685.11	1,500.00	185.11	
165—Library	538.52	300.00	238.52	
166—Publication Expenses (Bulletins)	2,013.33	4,000.00		1,986.67
167—Staff Insurance	186.46	400.00		213.54
168—Professional Services	368.73	500.00		131.27
169—Unallocated	690.47	2,000.00		1,309.53
170—Contingencies		2,500.00		2,500.00

171—Education and Information	3,661.66	5,000.00		1,338.34
250—Laboratory Materials and Supplies	3,497.67	10,000.00		6,502.33
300—Permanent Equipment	3,594.28	10,000.00		6,405.72
	24,209.42	45,000.00	458.29	21,248.87
LABORATORY OPERATION AND MAINTENANCE				
200—Mortgage Interest	78.75	200.00		121.25
202—Heat	1,031.36	1,200.00		168.64
203—Utilities	1,388.00	1,500.00		112.00
204—Janitor Wages	2,223.53	2,500.00		276.47
205—Janitor Supplies	148.12	300.00		151.88
206—Building Maintenance	1,341.15	1,500.00		158.85
207—Building Alterations	2,808.13	2,000.00	808.13	
208—Insurance	1,564.50	2,000.00		435.50
210—Contingencies	736.00	800.00		64.00
	11,319.54	12,000.00	808.13	1,488.59
COOPERATIVE RESEARCH				
1000—Pledged to June 30, 1949	6,541.00	6,800.00		259.00
1100—Pledged to October 31, 1949	7,500.00	13,800.00		6,300.00
	14,041.00	20,600.00	—0—	6,559.00
NAVY				
COMMITTEE EXPENSE				
N100—Travel and Meeting	403.95	2,000.00		1,596.05
STAFF SALARIES				
N150—Engineering and Tech. Staff	8,647.99	10,500.00		1,852.01
N151—Clerical Staff		1,000.00		1,000.00
N153—Part Time Staff and Contingencies		1,000.00		1,000.00
LABORATORY EXPENSES				
N160—Laboratory Expense—Staff	513.72	2,000.00		1,486.28
N250—Laboratory Materials	4,369.85	10,000.00		5,630.15
Laboratory Supplies	2,892.39	20,000.00		17,107.61
	16,827.90	46,500.00	—0—	29,672.10
TOTALS	$146,023.88	$222,500.00	$ 1,461.01	$ 77,937.13
UNBUDGETED DISBURSEMENT				
Advances to Employees Hospitalization	$ 82.60			

NEW BY-LAWS ADOPTED

President Stacey then called attention to the Proposed By-Laws, which are to replace the present Constitution, By-Laws, and Rules of the Society, and the Regulations Governing the Committee on Research. These revised By-Laws were approved and recommended to the Society for action at a Special Meeting held on November 21, 1949, in New York City. In accordance with the requirement of the present Constitution and the laws of the State of New York, the Proposed By-Laws have been presented to the membership of the Society (see Chapter 1378).

The first technical paper, Night-Air Cooling, by F. E. Giesecke (see Chapter 1379), was presented by Dean J. S. Hopper of Arlington State College, Arling-

ton, Tex. Dean Hopper expressed the regrets of Dr. Giesecke, who had been prevented from attending the meeting by the illness of his wife.

The next technical paper was Physiologic Examination of the Effective Temperature Index, by Nathaniel Glickman, Tohru Inouye, R. W. Keeton, M.D., and M. K. Fahnestock, which was presented by Professor Glickman (see Chapter 1380).

SECOND SESSION, TUESDAY, JANUARY 24TH, 9:30 A.M.

The second session was called to order by First Vice President Avery, who asked Mr. Worsham, chairman of the Board of Tellers, to report the results of the election, and the vote on the adoption of the proposed new By-Laws of the Society. The report follows:

REPORT OF TELLERS OF ELECTION

Ballot for Officers	Total
President, Lester T. Avery, Cleveland, Ohio	1141
1st Vice Pres., L. E. Seeley, Durham, N. H.	1141
2nd Vice Pres., Ernest Szekely, Milwaukee, Wis.	1141
Treasurer, Reg F. Taylor, Houston, Tex.	1132
Members of Council (Three-year Term)	
J. E. Haines, Minneapolis, Minn.	1083
J. W. James, Chicago, Ill.	1081
E. R. Queer, State College, Pa.	1070
G. B. Supple, Indianapolis, Ind.	1069
TOTAL BALLOTS RECEIVED	1206
TOTAL LEGAL BALLOTS	1143
INVALID BALLOTS	63
Scattering votes:	
President	2
1st Vice Pres.	1
2nd Vice Pres.	2
Council	8
Comm. on Res.	15
Committee on Research (Three-year Term)	
C. F. Boester, Lafayette, Ind.	1063
R. C. Cross, Chicago, Ill.	1075
R. S. Dill, Washington, D. C.	1062
A. J. Hess, Los Angeles, Calif.	1064
H. A. Lockhart, Morton Grove, Ill.	1078
One-year Term	
P. B. Gordon, New York	1084
Adoption of Society By-Laws	
For	1084
Against	50

Respectfully submitted,

Ira W. Wilke
Paul D. Close
Herman Worsham, *Chairman*

The first technical paper, Thermodynamic Criteria for Heat Pump Performance, by John F. Sandfort, was presented by the author (see Chapter 1381).

F. R. Ellenberger presented the paper, Evaluating Heat Pump Performance, by F. R. Ellenberger, A. B. Hubbard, W. R. Foote, F. Burggraf and J. J. Martin, Jr. (see Chapter 1382).

The next technical paper was Condensation on Prefabricated Walls, by E. R. Queer and E. R. McLaughlin, which was presented by Professor Queer (see Chapter 1383).

THIRD SESSION, WEDNESDAY, JANUARY 25TH, 9:30 A.M.

Second Vice President L. E. Seeley was the presiding officer at the third session. He called upon Prof. A. G. H. Dietz to present the paper, Solar Heating of Houses by Vertical South Wall Storage Panels, by Albert G. H. Dietz and Edmund L. Czapek (see Chapter 1384).

The next technical paper, Removal of Internal Radiation by Cooling Panels, by Merl Baker, was presented by the author (see Chapter 1385).

The last paper of the session, Baseboard Radiation Performance in Occupied Dwellings, by G. S. MacLeod and C. E. Eves, was presented by Mr. MacLeod (see Chapter 1386).

FOURTH SESSION, WEDNESDAY, JANUARY 25TH, 2:00 P.M.

Pres. A. E. Stacey, Jr. called the meeting to order and presented L. N. Hunter, chairman of the Committee on Research, who presented the 1949 Annual Report of the Committee on Research, followed by the report of the Director of Research.

ANNUAL REPORT OF COMMITTEE ON RESEARCH—1949

THE Committee on Research feels that this past year has been a successful one as far as Society research is concerned, but that this success is due partly to the cooperation and advice which it has received from several sources. The Council and Officers of the Society have given the Committee splendid cooperation and have been very helpful in offering valuable suggestions from time to time, which is greatly appreciated by the Committee on Research.

Finances: We refer you to the auditor's report and the Society's financial statement for information on Research finances.

The Committee wishes to acknowledge the splendid financial support received from industry during this past year. In 1949 there were 240 organizations which participated financially in our research program. C. A. McKeeman, formerly assistant to the president, who spent most of his time stimulating industry's interest and desire to participate in our research work, resigned last August. Since then, the Committee on Promotion of Research, under the chairmanship of J. E. Haines, has been carrying on with the fund raising responsibilities with the assistance of George Aubrey Hastings, the Society's public relations counsel, and Cyril Tasker, director of research.

We have no reason to feel that industry will not support our work in the future more extensively than in the past, if kept fully informed on our programs and our objectives.

Committee Activities: The Committee on Research held four meetings during the year and the Research Executive Committee held three. One of the things that is characteristic of the Committee on Research meetings is the excellent attendance

record of the committee members and the seriousness with which they assume their responsibilities. The chairman wishes to thank all of the committee members for the fine cooperation received during the year.

We now have 19 different Technical Advisory Committees, which have a total membership of 256 persons. These Committees held 33 meetings during the year. The total number attending these meetings was 265 members and 206 guests, including members of the Laboratory Staff.

The Committee on Research wishes to express its appreciation to all of the TA Committee members, and especially to the chairmen for their support and valuable contributions, advice and time which they have so freely given. The TA Committees have many responsibilities and we cannot emphasize too strongly the importance of their work.

Budget for 1950: The Research Committee submitted to Council several months ago a budget for the ensuing year, and at the October 17 meeting, Council approved a budget of $206,250 for 1950 research operations. Of this amount, $37,500 is for the project involving the Study of Human Calorimetry for the Navy, and the balance, $168,750, is for other Laboratory projects and cooperative research contracts.

Laboratory Staff: The Committee on Research wishes to acknowledge the excellent cooperation of the Laboratory staff under the able direction of Cyril Tasker, director of research. The interest shown by each staff member in his work and the teamwork that has been demonstrated by the staff as a whole have been very gratifying. The Committee recommended the reappointment of Mr. Tasker as director of research for the ensuing year, which recommendation has been approved by Council.

Publications and Papers: All research results are eventually made available to our members in published form, and in papers presented before the Society. They are published in the A.S.H.V.E. JOURNAL and TRANSACTIONS. Where it seems advisable to do so, they are also published in the form of research bulletins.

Professional societies as a rule have difficulty in establishing a publication policy that suits everyone, and this Society is no exception. The results of our membership survey a year ago clearly indicated that many of our members would like to have the results of research presented in a more easily readable form and in such a way that they can be readily used for practical application. The Committee on Research has the responsibility of seeing that research results are presented in a complete and scientific form in order that they be made available for future reference.

The application engineer is usually interested only in the final results expressed in a readily usable and condensed form. There is a question, however, as to whether preparation of the results in this form should be the function of the Committee on Research or of some other part of our Society's organization. THE GUIDE serves this purpose, but, of course, there is a period of time which elapses between the original publication of the results and the publication of THE GUIDE.

Research Policy: The Committee on Research has to work within the scope of the rules and regulations which have been laid down for it by the Society. However, within the scope of these rules and regulations there are many policies and interpretations that have to be decided. In order to maintain a consistent research policy, a Research Policy Subcommittee of the Research Committee has been formed, under the chairmanship of Prof. E. R. Queer, to study policy matters and make recommendations on the same to the Committee on Research for adoption.

The Committee on Research has been considering for some time ways of encouraging more of our members to participate in the planning of our research work. Because many of our Society members who are not on TA Committees have shown an interest in the work of those committees, we tried an experiment at the 1949 Annual Meeting in Chicago, and Society members were given an opportunity to attend the meetings of the TA Committees, whether they were members of the TA Committees or not. Previously, attendance was limited to Committee members and invited guests only.

Many of members took advantage of this opportunity and the reaction was so favorable that the same policy has been adopted regarding attendance at the TA Committee meetings held during the 1950 Annual Meeting. Consideration will be given to making this policy a permanent one.

Research Program: We did not spend quite as much money in 1949 as we expected we would, mainly for two reasons:

Although real progress has been made by the Laboratory Staff in the work involving the study of Human Calorimetry, we spent about $30,000 less on this project than we expected to. This was largely because it was found that much less expensive equipment could be used in carrying on this research work than was originally anticipated. This program is financed by the Office of Naval Research.

As most of you know, the business outlook last winter and early last spring was quite uncertain and it was felt advisable, therefore, to defer temporarily the expenditure of money for building two psychrometric rooms. However, we are definitely going ahead with these. Part of the research work for these rooms is on behalf of the studies for the TAC on Panel Heating and Cooling and part for the program proposed by the TAC on Sensations of Comfort. This is a new Committee appointed this year under the chairmanship of C. S. Leopold, and the scope of its activities will cover, primarily, studies of sensory reactions to environmental conditions at or near the optimum.

The Committee on Research continues to encourage projects with cooperating institutions. Contracts were completed with a number of them in 1949, and other proposals have been received and are under consideration. There is frequently quite a bit of work involved in setting up a cooperative project, and sometimes the progress may seem slow. Negotiations between the prospective cooperating institution and the Laboratory must be completed to a point where there is sufficient information available to permit the project to be referred to the proper TA Committee, so they may review it and advise the Committee on Research on what action should be taken on the proposal. The easiest kind of project to set up is one where the cooperating institution has the necessary facilities and qualified personnel to undertake a specific project in which the Society is particularly interested. Occasionally, however, an institution will write to the Society advising that they want to do some research work and ask us to assign them a problem and the necessary funds. It is difficult to do this if you do not know the personnel and the facilities that are available at the institution and, consequently, negotiations are sometimes extended over a greater period of time than normally would be the case.

One year ago the Committee on Research made a survey of our membership to determine the attitude and feelings of the Society membership toward our research program and its objectives. It was hoped that the results would serve as a valuable guide in formulating future research programs. The results of this survey were presented to the Society at the 1949 Semi-Annual Meeting, and were published in A.S.H.V.E. JOURNAL SECTION, *Heating Piping & Air Conditioning,* August 1949, p. 123. The results indicated that the current research program was pretty much in line with the thinking of our membership, but that more emphasis should be placed on the heat pump and on odors. Consequently, the TA Committee on Heat Pump has been quite active during this past year. One cooperative research project on this subject has been approved and others are under consideration.

The TAC on Odors has been quite active, and is giving consideration to this very complex problem and it is expected that a research program on odors will soon be under way. The scope of this committee involves the study of odor problems, including the formation, methods of measurements, and control of odors, and the physiolgical reactions to them.

The report of the director of research, which follows, will summarize for you the past year's work at the Laboratory and cooperating institutions and the activities of the TA Committees.

REPORT OF THE DIRECTOR OF RESEARCH

As reported by Cyril Tasker, director of research, steady progress was made in 1949 in most of the major projects on the Society's research program, among which were those concerned with heat gain through sunlit building surfaces; panel heating; certain parts of the overall problem of air flow and air distribution in rooms; air cleaning; and the combustion characteristics of fuel oils. At the eight institutions cooperating with the Committee on Research, investigations were under way on physiological reactions to atmospheric environment; air flow and air distribution; certain phases of the problem of cooling loads; and some parts of the research program in panel heating. Under a contract with the Office of Naval Research the Laboratory gave continuous attention to the problem of designing and constructing a calorimeter for the study of heat exchange between the human body and its surroundings.

Staff: Though there were no important changes in the Laboratory's engineering staff during 1949, the number and the variety of problems under investigation showed once more how important it is in any research organization to develop the team spirit and use the combined skills and ideas of the team in the preliminary phases of any investigation and from time to time as the work proceeds. At the year end the full-time staff numbered 24, of whom 12 were graduate scientific staff. Part-time clerical and technical staff, and the consulting services of members of the faculty of Case Institute of Technology, were also used during the year as needed.

Equipment and Property: Important additions were made to the library and several major items of laboratory equipment were added to a growing inventory of scientific apparatus available for laboratory or field projects. A number of important improvements were made to the property in 1949. Fire-resistant storage space was provided in the laboratory building for important laboratory papers, original data and records.

Accounting: During 1949 all research expenditures were made from Cleveland and all accounting carried on there. This system has worked with the utmost satisfaction and has made it possible to provide the Committee on Research and the Finance Committee at any time with accurate and up-to-date figures, not only on total income and expenditures but with detailed expenditures on each project on the program.

STUDIES AT CLEVELAND LABORATORY AND COOPERATING INSTITUTIONS

Only brief attention can be drawn to some of the main accomplishments in so extensive and varied a program. Research bulletins, or research papers, form permanent records of the studies carried on under the direction of the Committee on Research. The major projects are outlined as follows:

Air Flow and Distribution: To realize more effectively its long-range objective of establishing reliable fundamental data at the same time as it investigates important practical problems, studies under the Technical Advisory Committee on Air Distribution were divided by the committee into three main groups, dealing respectively with air flow in ducts and fittings, room air distribution, and field practices. Two outline charts, one on duct flow and one on room air distribution, which were first prepared in 1946, were revised in order to classify design data and application problems in the light of present knowledge.

Air Flow in Ducts and Fittings: At Michigan State College, under Dean L. G. Miller, studies are under way on the resistance to the flow of air offered by take-offs from a rectangular main to rectangular branches involving a 90-deg turn. Three types of take-offs are being studied: (*a*) Where main expands, in width only, to include total width of branch and continuation of main, and branch includes a standard 90-deg ell; (*b*) Main is decreased in width to that of continuing main while branch is taken off from sloping side of decreaser with a fitting and a 45-deg angle; and (*c*) The so-called *tailormade* fitting where the 90-deg ell is built into main.

At the Laboratory a survey is under way of available information on losses in through-flow duct bends. A bibliography of over 150 references has so far been com-

piled; the most important of these are being carefully studied. By following this procedure it is hoped that a sufficient amount of data can be compared on a common basis to reduce the confusion caused by the wide variations in test results from various investigators. The survey is designed to point out those factors which require additional experimental investigation.

Room Air Distribution: Studies have continued at the Laboratory, at Case Institute of Technology and at Kansas State College.

At the Laboratory a very comprehensive survey and critical analysis of available information on the fundamentals of ventilation jet behavior have been completed and are being prepared for publication. Experimental studies on jets discharging horizontally into a large free space are under way, initially using a 6-in. round nozzle. Jet outlet temperatures ranging from 50 deg below the surrounding temperature to 80 deg above can be provided with velocities ranging from 100 fpm to 6000 fpm. Turbulence can be varied over a wide range. The present studies with chilled jets are covering a region where no good data exist at present.

Most studies of air distribution depend greatly on the ability to measure low air velocities with reasonable accuracy and rapidity. Since suitable commercial instruments were not available, the Laboratory undertook the development and construction of a heated-thermocouple anemometer capable of giving reproducible results within the range required. A paper was prepared on this work and submitted for publication, to stimulate the interest of others in the problem of measuring low air velocities.

Case Institute of Technology: Under the direction of Prof. G. L. Tuve numerous tests have been made to determine the extent of similarity in the behavior of air streams projected from a free opening, a grille or register of the commercial type with a percentage free area ranging from 55 to 85, and a perforated panel with a free opening ranging from 3 to 15 percent. It is believed that they may behave in a similar manner after the jet coalescence has been taken into account. It is also thought that flow data on all types of outlets can be correlated to the extent that the approximate throw and spread of the stream can be predicted by the same method for all devices. However, it is evident, Prof. Tuve reports, that there are still unknown factors calling for further research.

Studies with slotted type outlets are continuing. A paper[1] on this subject was presented at the 1949 Semi-Annual Meeting in Minneapolis.

Kansas State College: Studies of downwardly projected warm air streams are being made under the direction of Prof. Linn Helander and with the active support of the *Industrial Unit Heater Association.* Experimental work was extended to cover temperatures and velocities along the vertical axis of streams projected from 2-in. and 4-in. diameter nozzles. Additional mathematical analysis has resulted in formulas for the maximum downward blow of other types of idealized streams than that reported in an earlier paper[2].

With the cooperation of members of *I.U.H.A.* in the loan of equipment and provision of additional funds, test equipment has now been installed at Manhattan that will permit the study of streams projected downward from orifices between 6-in. and 20-in. in diameter and from heights up to 35 ft. Studies with ambient air and heated air are planned for the early part of 1950.

Physiological Research—Human Calorimetry (O.N.R. Contract): In the 1948 report brief mention was made of the scope of the investigation under way under a contract with the Office of Naval Research and with the cooperation of the *Naval Medical Research Institute.* A special advisory committee established the design criteria for the calorimeter and the auxiliary equipment. The most important decision was that the calorimeter shell, should be designed as a double-walled rectangular compartment 32-in. square in cross section by 84-in. long with openings provided for respiratory and ventilatory air ducts and for the visual observation of the subject.

The special heat metering layer, originally suggested by Dr. T. H. Benzinger, was found to require additional development before it could be applied satisfactorily to large areas. This development and accelerated-use test procedures were completed in

[1] A.S.H.V.E. RESEARCH REPORT No. 1366—Air Streams from Perforated Panels, by Alfred Koestel, Philip Hermann and G. L. Tuve (A.S.H.V.E. TRANSACTIONS, Vol. 55, 1949, p. 283).

[2] A.S.H.V.E. RESEARCH REPORT No. 1327—Downward Projection of Heated Air, by Linn Helander and C. V. Jakowatz (A.S.H.V.E. TRANSACTIONS, Vol. 54, 1948, p. 71).

November. All the layer sections for the calorimeter shell are now complete and ready for installation. Special equipment to provide the necessary bonding temperature is under construction.

One of the most difficult problems facing the Laboratory staff has been to design the fluid flow circuits in such a way as to give the required flexibility and still retain the very close temperature control suggested as essential to the operation of the calorimeter.

The satisfactory operation of the ventilatory air circuit depends almost entirely on the design and construction of so-called *plate meters*. To obtain performance data a pilot plate meter unit was constructed and subjected to a long series of tests from which data were secured for use in the final design now awaiting approval.

All operating costs on this project are covered from funds made available by the Office of Naval Research.

Environmental Effects: Studies at the College of Medicine, University of Illinois, on the physiological adjustments of healthy and physically impaired persons to sudden changes in environment were continued throughout the year. During 1949 financial support of these studies was augmented by means of a grant-in-aid of the National Institutes of Health, U. S. Public Health Service, and by funds provided by the Association of American Railroads.

A paper[3] reported the results obtained in a study designed to compare, by physiological measurements, the response of healthy subjects to two different ambient environments having the same effective temperature. Relative humidities of 30 percent and 80 percent and effective temperatures in the zones of body cooling, thermal neutrality, and evaporative regulation were selected. Studies were made with the subjects in approximate equilibrium with the environment. The results suggest, it is believed, that the present ET index is adequate for subjects in the transient state except that some correction is needed at the higher temperatures (ET's of 82 and above). For subjects in a steady-state it is suggested that the present ET index places too much emphasis on the influence of relative humidity for the range of ambient conditions studied, (dry bulb temperatures from 60 F to 97 F; ET's from 59½ to 82½ deg). These studies are continuing with grossly obese, lean and emaciated subjects and those with other physical impairments.

Considerable interest is being displayed in many parts of the world in questions of thermal comfort in temperate and tropical climates. The Society is cooperating in a rather extensive exchange of information between the various investigators in this field, some of whom are corresponding (foreign) members of the TAC on Physiological Research.

Under the sponsorship of the TAC on Sensations of Comfort important changes have been made on the Comfort Chart to be published in the HEATING, VENTILATING, AIR CONDITIONING GUIDE 1950 to reflect the fact that both laboratory research and field observations over the past several years indicate that the optimum dry bulb temperature for winter is about 75 F and that this seems to hold over a fairly wide range of relative humidities.

The calorimeter room now under construction at Cleveland will enable the Laboratory once again to engage directly in studies on sensory and physiological reactions of persons to environmental conditions.

Heat Flow and Heat Transfer: Studies on the effects of low velocity air movements on heat transfer from a smooth surface were completed; the results are being prepared for publication.

Heat Transmission Through Glass: The experimental work on glass block was continued through June 1949, solar transmittance data being obtained for nine panels which included three sizes of block and six patterns. Tests were also made to determine the heat gain by convection and radiation from the indoor surface of the panel and to determine overall coefficients under steady-state natural weather conditions. A paper[4] was presented at the 1949 Semi-Annual Meeting. Further analysis is being made of heat flow by radiation and convection.

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[3] A.S.H.V.E. RESEARCH REPORT No. 1380—Physiologic Examination of the Effective Temperature Index, by Nathaniel Glickman, Tohru Inouye, R. W. Keeton and M. K. Fahnestock (See p. 51).

[4] A.S.H.V.E. RESEARCH REPORT No. 1374—Solar Energy Transmittance of Eight-Inch Hollow Glass Block, by George V. Parmelee and Warren W. Aubele (A.S.H.V.E. TRANSACTIONS, Vol. 55, 1949, p. 435).

Experimental work on configurated glass was conducted between July and October; the results are now being prepared for presentation.

In the 1948 report it was mentioned that a convection-compensated radiometer had been designed and constructed at the Laboratory to measure incoming radiation from all sources. During 1949 it was subjected to considerable development, testing and use. After wind tunnel tests it was mounted on the calorimeter, and has been used in all tests since then to measure the long wave length radiant energy received from the outdoor surroundings. A small blower was later added to produce a more constant convection effect. Calibrations carried out indoors with a standard black body indicated that the error of the instrument was of the order of 3 percent for the usual range of measurements. Tests with the instrument mounted on a vertical wall show that early and late in the day the outdoor surroundings, including the sky, contribute substantially less radiation than a black body radiating at air temperature. The magnitude of this difference is significant, and in tests made at the Laboratory amounted to 20 Btu per (hr) (sq ft). Near noon, due to solar heating, the surroundings emit nearly as much radiation as a black body would, radiating at air temperature.

Design Data on Heat Flow Through Glass: The true significance of the results of the extensive investigations made in the past three years on heat transmission through glass will not be realized until the experimental data are translated into design data and put into such a form that they may be readily used by design engineers and others interested in calculations of heat flow through sunlit building surfaces. Aided by the advice of a special subcommittee of the TAC on Cooling Load and TAC on Heat Flow Through Glass, the Laboratory staff have made marked progress in this phase of the program.

Design heat flow tables for selected solar and weather conditions for six patterns of 8-in. glass block have been prepared. Additions have been made to design data for flat glass. Tables have been prepared giving overall coefficients for flat glass and glass block, for stated design conditions. A paper reviewing the Society's work in this field was presented by G. V. Parmelee before the *American Ceramic Society* in Cincinnati in April 1949.

It is gratifying to be able to report substantial technical and financial support of these programs during 1949. The work to date has all been fundamental in character and has involved no *testing* of any commercial products whatsoever. The support given to these studies over the past few years indicates what can be accomplished if a long-range research program is soundly organized and developed, and adequately presented to interested segments of this and allied industries. It is believed that the design data, when fully developed, will indicate how important these studies have been to the air conditioning industry.

Cooling Loads—Periodic Heat Flow: At Cornell University Prof. C. O. Mackey is continuing his work on the sol-air thermometer under a cooperative agreement with the Society. Studies are under way to discover the effect of different materials of construction and different sizes of exposed surface on the readings given by the instrument. At Tulane University, also under a cooperative agreement, sol-air temperature data have been developed for New Orleans. These data will add values for a humid area where the effect of water vapor in the atmosphere may be significant.

Heat Pump: The TAC on the Heat Pump held its organization meeting in January 1949 and began work on the determination and evaluation of satisfactory heat sources for heat pump applications in temperate zones. It has maintained close contact with all other known bodies interested in this general field, and committee members have prepared outlines covering existing information, investigations and suggestions for research on the following heat sources: air, earth, water, solar energy. It is planned to issue a composite review of the information gathered to date, to act as background material for future studies.

Under a cooperative research contract with the Society, Dr. R. C. Jordan of the University of Minnesota is making a study on the availability of solar energy as a potential heat source for the heat pump, the design of solar radiation absorbers and the use of storage media.

Panel Heating and Cooling: In the latter part of 1947, the Society undertook a long-range program of research in panel heating and cooling to *develop data to permit the proper calculation and design of panel heating and cooling systems, with present emphasis on applications for human comfort.* The Technical Advisory Committee on

Panel Heating and Cooling, which was formed to assist the Committee on Research and the Laboratory staff in directing the program and interpreting the results, divided the investigation into four divisions, three of which were assigned to separate groups.

The broad outlines of the program were given in the Annual Report of the Committee on Research for 1947 (see 1948 TRANSACTIONS, p. 42). Progress was reviewed in the Annual Reports of the Committee on Research for 1948 (see 1949 TRANSACTIONS, p. 11) and 1949 (see p. 16), and a brief report of some of the work was given at the 55th Annual Meeting in Chicago in January, 1949[5].

Under Group A, which was concerned with heat distribution within and behind the panel, the design and construction of test equipment was begun early in 1948, and laboratory experiments on concrete panels were carried on from the Fall of 1948 to the Summer of 1949.

Experimental studies have been made with three concrete slabs, and the data thus obtained verified and the range of application extended, by means of an electrical analogue.

Analytical studies have been made of heat flow in a concrete panel when pipes are spaced uniformly and buried within a solid slab. Field tests have been initiated on heat flow from a concrete slab to the ground, as influenced by the type and location of insulation below the slab.

Tests have begun on plaster panels, with the tubing both below and above the metal lath.

Progress has been made in the preparation of a method of design for panel heating suitable for use in THE GUIDE. The bibliography of articles and references on panel heating and cooling has been brought up to date to the end of 1949, and issued as a supplement to the bibliography prepared and distributed in 1948.

The tests made on concrete slabs covered three slabs of different thicknesses and with different depths of bury of the pipe. Pipe spacings were on 4-in. centers, thus enabling tests to be made at 4-in., 8-in., 12-in. and 16-in. centers. By using special equipment designed and constructed at the Laboratory, it was possible to set the test conditions so that the heat release from the upper surface of the slab was approximately 0, 25, 50, 75 and 100 percent of the heat supplied, the remainder being given off by the lower surface. Though most of the tests were made with the slab surface bare, some tests were made with the slab surfaces covered with carpet and carpet padding, with several thicknesses of insulation and with asphalt tile.

Fairly complete reports covering the various studies made to date have been prepared for committee study. It must be emphasized that the information secured to date is not in itself adequate for design purposes but forms an important part of the data required for the development of simple design procedures. The validity of using a thermal test system has been proved, the electrical analogue has been shown to be a convenient means of extending the range of thermal observations, the test data have confirmed the fundamental theory developed by mathematical analysis for uniformly-spaced pipes buried within a solid slab, provided the effects of the slab surface temperature-wave are negligible, or in other words, where the slab surfaces may be considered to be isothermal for practical applications. Some work has been done on the extension of experiment and theory to the case of slabs where the slab surfaces cannot be considered isothermal. Other studies have been commenced for the case of the slab on earth, where the pipes are in the slab-earth interface.

At Columbia University, under a cooperative agreement with the Society, Prof. Carl F. Kayan completed his study of the effect of idle tubes on the heat flow pattern in a concrete slab. Using electrical analogy methods, he not only confirmed the results obtained in the thermal studies made at the Laboratory but obtained interesting data on surface and slab temperature distribution.

Field Study: For the field study four frame, one-story, five-room houses of identical construction, except for under-floor insulation, were instrumented for underfloor and floor temperature and heat flow distribution studies. All houses face the same direction and are built on adjacent lots. Types of insulation are hollow tile and glass foam with some variations in application.

Thermal measurements are being currently made and will continue through the

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[5] Performance of Heating Panel Studied (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping & Air Conditioning*, April 1949, pp. 112-114).

present heating season. A mechanical soil analysis has been made to assist in establishing soil thermal conductivity at the site.

Calorimeter Room: To provide facilities for the study of heat flow from the panel to the space, plans and drawings were made early in 1949 for a calorimeter room about 16 ft × 12 ft with an 8 to 10-ft ceiling. These plans were extensively modified in the early summer to accommodate the needs of the TAC on Sensations of Comfort. Plans approved late in the year are for a room 24 ft long and 12 ft wide with a movable ceiling of 12 ft maximum height. This room will be located in the upper Laboratory, with heavy service equipment located directly below in the lower Laboratory. Preliminary construction has begun, and many of the materials are now on order; the type of aluminum radiant panel selected is one that has been tried out commercially with considerable success.

The Laboratory has urgently needed facilities of this type ever since it was moved to the present permanent quarters. The staff anticipate that these facilities, when completed, will be kept in constant use for many different types of investigation.

Industry Support: The panel heating research program has received substantial support in 1949 from several trade associations as well as from a number of industrial organizations. The cost of the studies made in 1949 has, however, been borne almost equally by earmarked contributions and by general Society funds available to the Committee on Research. A substantial part of the funds contributed in 1949 has been carried over to the 1950 fiscal period in which expenditures, especially on the calorimeter room, are likely to be heavy.

Air Cleaning: As part of the plan to develop acceptable test procedures, the Laboratory has made some studies on test dusts. A subcommittee on test dusts was set up to advise and assist in this part of the problem. In order to specify the characteristics which standard test dusts should possess, it seemed desirable to study the types of air contamination found in the atmosphere in different parts of the country and relate these to the types of material which should be used in accelerated-use tests. The Laboratory has played its part in such a study, and we are indebted to the Bell Telephone Laboratories for the loan of the four sampling units and to the three filter manufacturers who are cooperating with us in the study.

At the Laboratory various methods for determining size and particle shape characteristics of dusts have been studied and techniques developed. Problems in velocity and dust distribution in the test duct have been overcome and satisfactory sampling positions tentatively established.

A number of important questions remain to be answered before the Laboratory will be in a position to suggest test procedures. Similar problems exist in other countries and close touch is being maintained with other workers in the field.

A paper[6] being presented at the 56th Annual Meeting covers studies made a few years ago under a cooperative research agreement between the University of Minnesota and the Society and hitherto unreported.

Oil-Heat Institute: In cooperation with the *Oil-Heat Institute of America* Engineering Committee and its research subcommittee, a third investigation was made during 1949 to determine to what extent the combustion reference test unit might be used as a means of rating the burning quality of domestic fuel oils.

Instrumentation of the unit was improved with the addition of an air supply metering system and the use of a light reflection meter to evaluate the density of smoke deposit on smoke meter filter papers.

Test results with six oils of widely differing properties indicated that the unit was capable of distinguishing the respective smoking tendencies with reasonable reproducibility. Good correlation was obtained between the smoking tendency of these oils and several common indices of combustion. Turbulence of air supply and means of ignition were found to have a pronounced effect upon CO_2 as well as upon smoke density. It was concluded that in order to establish the true significance of the results obtained with a unit of this nature it would be necessary to investigate fundamental principles of fuel atomization, ignition, air-fuel mixing, and particle size in smoke indication.

A report has been made to the *O.H.I.* Engineering Committee. The results of the

[6] A.S.H.V.E. RESEARCH REPORT No. 1391—Resistance Gradients through Viscous Coated Air Filters, by F. B. Rowley and R. C. Jordan (see p. 237).

three investigations made to date are being prepared for presentation as a research paper before the Society.

The Guide—Psychrometric Chart: (a) In line with the request of the Guide Publication Committee that the Laboratory assist in the development of a new psychrometric chart, H. B. Nottage, research associate at the Laboratory, has cooperated with a special subcommittee in preparing first a skeleton chart, and finally a large-scale master chart. This chart is now being retraced. A memorandum has been prepared describing the background of the chart, the principles upon which it is drawn and the proposed manner of its use. The chart and memorandum will shortly be presented to a representative group of Society members for criticism and discussion as a prelude to presentation at the 1950 Semi-Annual Meeting.

(b) Many members of the Committee on Research, and of the TA Committees, as well as several staff members, have assisted in the revision of a number of Guide chapters. The TAC on Sound Control was responsible for the rewriting of the Chapter on Sound Control in THE GUIDE. Members of the TAC on Industrial Ventilation have been responsible for revision or rewriting of the Chapters on Air Cleaning, Air Contaminants, Industrial Air Conditioning and Industrial Exhaust Systems in THE GUIDE. The director of research served as an ex-officio member of the Guide Publication Committee to insure close liaison between it, the Committee on Research and the various technical advisory committees.

Industrial Ventilation: In addition to assisting the Guide Publication Committee, the TAC on Industrial Ventilation has conducted a comprehensive survey of industrial spot cooling installations and practices by means of a questionnaire sent to over 400 organizations. The returns were gratifying and replies are now being analyzed.

Odors: During 1949 attention was directed to the problem of odors in occupied spaces by setting up a Technical Advisory Committee on Odors to *study odor problems; including the formation, methods of measurement and control of odors, and the physiological reactions to them.* Two meetings were held and plans made to develop active studies in this field in 1950. It is evident that the problem is an important one, but there are some wide differences of opinion as to the best methods of attack.

Sorbents: The TAC on Sorbents is at work on a recommended method for the testing and rating of sorption-type dehumidifiers.

RESEARCH PUBLICITY AND PROMOTION

Senior staff members have participated in inter-society activities with *A.I.Ch.E., A.I.E.E., A.S.M.E., A.S.R.E.,* and the *I.E.S.* Close liaison has been maintained with workers in our own and allied fields here and abroad.

Five members of the staff spoke on the research program before the Northern Ohio Chapter in May. G. V. Parmelee spoke at two chapters of the Society during 1949 as well as at the meeting of the *American Ceramic Society.* The director of research visited 11 chapters in 1949 and also spoke before the United Nations Scientific Conference on the Conservation and Utilization of Resources, the *National Association of Master Plumbers* and the 18th Annual Forced Warm Air Conference at Michigan State College.

Visits to chapters by members of the Laboratory staff appear to serve perhaps better than most other media to bring before the membership the purpose and significance of the Society's research program. Though substantial progress has been made in research publicity and promotion during 1949, it is and must be a continuing program, deserving of careful planning and the use of a variety of media. The director and other Laboratory staff members cooperated with the Society's Public Relations Counsel in the preparation and distribution of a booklet entitled *Research—Investment in Progress.*

ACKNOWLEDGMENTS

On behalf of the Laboratory staff the writer takes this opportunity to express sincere appreciation for the encouragement and support given to it by the Society Officers and Council, and especially by the Chairman and members of the Committee on

Research and the members of the various Technical Advisory Committees. Although not all was accomplished that was hoped for at the beginning of the year, the year end saw research activities on a firm footing, both technically and financially. The staff look forward with confidence to 1950 in the belief that the work in which they are engaged will bring substantial benefits not only to the heating, ventilating and air conditioning industry but also to the general public.

The writer expresses thanks to all Laboratory staff members for their loyal service and unfailing cooperation.

President Stacey asked Mr. Tasker to introduce the subject of Panel Heating Research being carried on at the Research Laboratory. Mr. Tasker stated that the report would be presented in two parts, the Analytical Development, to be covered by H. B. Nottage, and the Testing Program and Results of Research, to be reported by C. M. Humphreys.

Prof. Carl F. Kayan of Columbia University Engineering Center gave a brief report of the cooperative research work which had been completed, using the Electric Analogger for the prediction of the steady-state heat transfer and internal temperature distribution for a radiant heat panel with different tube spacings under the following conditions: horizontal concrete slab containing ¾ in. heated imbedded tubes, with bottom cover of 5 in. and top cover of 3½ in., and also ¾ in. (assuming constant top and bottom airside conductances).

The next technical paper, Effect of Panel Location on Skin and Clothing Surface Temperature, by L. P. Herrington and R. J. Lorenzi (see Chapter 1390), was presented by Dr. C.-E. A. Winslow.

The meeting was adjourned at 4:30 p.m.

FIFTH SESSION, THURSDAY, JANUARY 26TH, 10:00 A.M.

The last session was called to order by Pres. Alfred E. Stacey, Jr., who asked Prof. R. C. Jordan to deliver the paper, Resistance Gradients Through Viscous Coated Air Filters, by F. B. Rowley and R. C. Jordan (see Chapter 1391).

The next technical paper, Fitting Losses for Extended-Plenum Forced Air Systems, by H. H. Korst, N. A. Buckley, S. Konzo and R. W. Roose (see Chapter 1392).

The final technical paper of the session, Vaneaxial Fan Fundamentals, by Raymond Mancha, was presented by the author (see Chapter 1393).

President Stacey expressed the appreciation of the Society to all of the authors of technical papers, and to those who had participated in the discussions.

I. W. Cotton, Chairman of the Committee on Resolutions, presented the following report, which was unanimously adopted.

RESOLUTIONS

The 56th Annual Meeting of the A.S.H.V.E., January 23 to 26, 1950, has been outstandingly successful and exceptionally pleasant.

Whereas: This circumstance did not just happen, it was the direct result of careful planning and hard work. We who have thoroughly enjoyed the meeting owe marked gratitude to those who conceived and then executed the various activities and programs making up the impressive schedule we are now completing, *Therefore,*

BE IT RESOLVED: That our thanks be expressed—

To those resolute Society Members, beginning with our beloved Dr. Giesecke and later spearheaded by C. Rollins Gardner who insisted persistently through lo! these many years that we break all tradition and hold this Annual Meeting in the great Southwest.

To the membership of the Texas Chapters whose superb hospitality has proven that the prognostications of Mr. Gardner and his cohorts which might have been considered over-optimistic boasts were in reality masterpieces of understatement.

To the gracious distaff side of our hosts' welcoming organization for entertaining our visiting Ladies so delightfully;

To the National and Chapter Officers and staffs who organized all arrangements so efficiently and provided such excellent meetings, tours and diversions;

To the large number of patient, enthusiastic and devoted searchers for knowledge who toiled in laboratory and field to develop new data, to the authors who wrote and the speakers who presented the papers based on such data at our interesting and informative technical programs; also to the commentators whose questions, amplifications and criticisms livened the sessions and contributed to their educational value;

To the Dallas hostelries for sheltering us comfortably and also for nourishing us against the strenuous hazards of the countless breath-taking dashes from Adolphus to Baker and likewise from Baker to Adolphus;

To the Weather Bureau Meteorologist who gallantly came to the front for our hosts by ordering a liberal sprinkling of June in January—thereby quieting the fears and consolidating the front of supreme confidence which graced the personality of Mr. Gardner in extolling all the virtues of his home town, except the weather, while preparing for the Meeting;

And finally, in order to beat the Texans to it, we had better proclaim our own astuteness—so unmistakably demonstrated by our flocking in droves from far and wide to come, see and be conquered by this great State of Texas—in numbers large enough, in fact, to enable Dallas to break many A.S.H.V.E. records and at the same time prove to the loyal sons of Texas that her greatness is fully shared by 47 friendly sister States which, joined by Canada, Mexico, France and Sweden, have sent so many Society members to enjoy her eminent hospitality.

Respectfully submitted,

THE RESOLUTIONS COMMITTEE

I. W. COTTON, *Chairman*, Indianapolis, Ind.
E. K. CAMPBELL, Kansas City, Mo. R. A. SHERMAN, Columbus, Ohio

INSTALLATION OF OFFICERS

The Society Officers elected for the year 1950, and inducted at the final session of the Annual Meeting in Dallas, were: *President*—Lester T. Avery, Cleveland, Ohio; *First Vice President*—L. E. Seeley, Durham, N. H.; *Second Vice President*—Ernest Szekely, Milwaukee, Wis.; *Treasurer*—Reg F. Taylor, Houston, Tex.

Four new Council members were elected as follows: John E. Haines, Minneapolis, Minn.; John W. James, Chicago, Ill.; E. R. Queer, State College, Pa.; G. B. Supple, Indianapolis, Ind.

At the request of President Stacey, S. H. Downs, Kalamazoo, Mich., past president of the Society, assumed the chair and took charge of the installation assisted by Walter L. Fleisher and Prof. G. L. Tuve.

As there was no further business, President-Elect Avery declared the 56th Annual Meeting adjourned.

PROGRAM 56th ANNUAL MEETING

Hotel Adolphus and The Baker Hotel, Dallas, Tex.

January 22-26, 1950

Sunday—January 22

10:00 a.m. RECEPTION & REGISTRATION (Baker—*Mezzanine Lounge*)
10:00 a.m. Committee on Research (Adolphus—*Parlor B*)
10:30 a.m. Council (Adolphus—*Parlor A*)
10:30 a.m. Committee on Admission and Advancement (Baker—*Room 4*)
3:00 p.m. Ladies Get-Together Tea (Baker—*Texas Room*)
Hostesses: Mmes. E. T. Gessell and J. A. Ray
4:30 p.m. Chapter Delegates (Adolphus—*Parlor D*)
8:00 p.m. TAC on Panel Heating and Cooling (Adolphus—*Parlor B*)
8:00 p.m Open Forum on A.S.H.V.E. By-Laws (Baker—*Room 1*)

Monday—January 23

8:30 a.m. REGISTRATION (Baker—*Mezzanine*)
8:30 a.m. RECEPTION AND INFORMATION (Adolphus—*Lobby*)
9:30 a.m. BUSINESS SESSION (Baker—*Ballroom*) Pres. Alfred E. Stacey, Jr. presided
Call to Order
Greetings by P. N. Vinther
Response by Alfred E. Stacey, Jr., President
Reports of Officers and Council
Night-Air Cooling, by F. E. Giesecke
Physiologic Examination of the Effective Temperature Index, by Nathaniel Glickman, Tohru Inouye, R. W. Keeton, M. K. Fahnestock
Amendment of By-Laws
10:30 a.m. Ladies Assembly (Baker—*Lounge*)
Mrs. T. H. Anspacher, *General Chairman*
12:15 p.m. WELCOME LUNCHEON (Adolphus—*Ballroom*)
Toastmaster: Stanley Foran
Speaker: Jeff Williams
2:00 p.m. OPENING OF EXPOSITION at State Fair Grounds by R. L. Thornton, President, State Fair of Texas and Chairman of Board, Merchantile National Bank, and The A.S.H.V.E. Society Officers
2:00 p.m. TAC on Air Distribution (Baker—*Room 2*)
2:00 p.m. TAC on Cooling Load (Baker—*Room 3*)
2:00 p.m. TAC on Insulation (Baker—*Room 4*)
2:30 p.m. Chapter Delegates (Adolphus—*Parlor D*)
2:30 p.m. Ladies Discussion of Fine Jewels by Joseph Linz (Baker—*Lounge*)
Hostesses: Mmes. W. R. Barbeck and J. P. Dillard
4:00 p.m. TAC on Sensations of Comfort (Baker—*Room 6*)
7:00 p.m. CHUCK WAGON DINNER—Entertainment—Dancing (Foods Building, Fair Park)

The Foods Building at Fair Park, where the Chuck Wagon Dinner was held, adjoined the Exposition building. The Chuck Wagon was open from 7:00 to 11:00 p.m., to accommodate members visiting the Exposition. Southwestern entertainment was presented throughout the evening. There was an exhibition of square dancing and audience participation in square and ballroom dancing.

Tuesday—January 24

9:00 a.m. REGISTRATION (Baker—*Mezzanine*)

9:30 a.m. TECHNICAL SESSION (Adolphus—*Ballroom*) First Vice Pres. Lester T. Avery presided

Report of Tellers of Election

Thermodynamic Criteria for Heat Pump Performance, by John F. Sandfort

Evaluating Heat Pump Performance, by F. R. Ellenberger, A. B. Hubbard, W. R. Foote, F. Burggraf, J. J. Martin, Jr.

Condensation on Prefabricated Walls, by E. R. Queer and E. R. McLaughlin

10:00 a.m. TAC on Cooling Load and Heat Flow through Glass (Adolphus—*Parlor E*)

12:30 p.m. LUNCHEON MEETING (Adolphus—*Roof Garden*, 15th Floor) M. F. Blankin presided

Speaker: K. C. Richmond, Editor, Coal-Heat Magazine

Subject: The Human Equation—A Public Relations Problem in Heating and Air Conditioning

12:30 p.m. Ladies Luncheon and Style Show (Baker—*Mural Room*)

Hostesses: Mmes. D. C. Kieffer and H. G. Clark, Jr.

2:00 p.m. Chapter Delegates (Adolphus—*Parlor D*)

2:00 p.m. TAC on Heat Pump (Baker—*Room 4*)

2:00 p.m. TAC on Sound Control (Baker—*Room 3*)

2:00 p.m. TAC on Heat Flow through Glass (Baker—*Room 5*)

2:30 p.m. Committee on Code for Testing and Rating Heavy Duty Furnaces (Adolphus—*Danish Room*)

6:30 p.m. Past Presidents' Dinner (Adolphus—*Parlor D*)

Wednesday—January 25

9:00 a.m. REGISTRATION (Baker—*Mezzanine*)

9:30 a.m. TECHNICAL SESSION (Adolphus—*Ballroom*) Second Vice Pres. Lauren E. Seeley presided

Solar Heating of Houses by Vertical South Wall Storage Panels, by A. G. H. Dietz and Edmund Czapek

Removal of Internal Radiation by Cooling Panels, by Merl Baker

Baseboard Radiation Performance in Occupied Dwellings, by G. S. MacLeod and C. E. Eves

1:45 p.m. Leave Baker Hotel—Ladies Theater Party—Margo Jones

Presented: "My Granny Van"

Hostesses: Mmes. G. A. Linskie and V. C. Kneese

2:00 p.m. TECHNICAL SESSION (Adolphus—*Ballroom*) Pres. Alfred E. Stacey, Jr. presided

Report of Committee on Research, by L. N. Hunter, *Chairman*

Panel Heating Research Report on Heat Flow Within A Concrete Panel

Experimental Techniques and Results, by C. M. Humphreys

Heat Flow Analysis For Buried Pipes, by H. B. Nottage

Electric Analogger Studies on Panels, by Carl F. Kayan

Effect of Panel Location on Skin and Clothing Surface Temperature, by L. P. Herrington and R. J. Lorenzi

4:30 p.m. Organization Meeting of 1950 Committee on Research

7:00 p.m. ANNUAL BANQUET (Baker—*Ballroom*)
Toastmaster: C. Rollins Gardner
Presentation of Memory Book to Prof. George L. Tuve by Past Pres. W. L. Fleisher, A.S.H.V.E.
Presentation of Past President's Emblem to Alfred E. Stacey, Jr., by Past Pres. W. L. Fleisher, A.S.H.V.E.
Presentation of F. Paul Anderson Medal to Dr. C.-E. A. Winslow, by Alfred E. Stacey, Jr., President, A.S.H.V.E.
Speaker: Dr. Umphrey Lee, President, Southern Methodist University.
Subject: The Fearful Era

Thursday—January 26

9:00 a.m. REGISTRATION (Baker—*Mezzanine*)

10:00 a.m. TECHNICAL SESSION (Adolphus—*Ballroom*) Pres. Alfred E. Stacey, Jr. presided
Resistance Gradients Through Viscous Coated Air Filters, by F. B. Rowley and R. C. Jordan
Fitting Losses for Extended-Plenum Forced Air Systems, by H. H. Korst, N. A. Buckley, S. Konzo, R. W. Roose
Vaneaxial Fan Fundamentals, by Raymond Mancha
Unfinished Business
New Business
Resolutions
Installation of Officers
Adjournment

12:30 p.m. Nominating Committee Luncheon (Adolphus—*Parlor F*)

1:30 p.m. COUNCIL MEETING (Baker—*Banquet Room 2*)

COMMITTEE ON ARRANGEMENTS

C. ROLLINS GARDNER, *General Chairman*

Vice Chairman
G. A. LINSKIE

Honorary Chairmen
F. E. GIESECKE REG F. TAYLOR

Banquet—M. L. Brown, *Chairman;* J. P. Dillard, J. R. Dowdell, W. E. Frost, E. J. Stern.

Entertainment—R. E. Allison, *Chairman;* A. B. Carter, V. C. Kneese, James O'Connor, Jr., A. A. Stone, J. L. Tye.

Exposition—Oslin Nation, *Chairman;* W. R. Barbeck, S. S. Brandt, N. R. Collins, Jr., F. L. Gray, Jr., R. G. Lyford.

Finance—E. T. Gessell, *Chairman;* J. B. Lowe, J. F. Marshall, R. R. Matthews.

Ladies—T. H. Anspacher, *Chairman;* R. R. Brient, Jr., R. A. Mixon, J. D. Poythress, *Assisted by:* Mmes. T. H. Anspacher, J. P. Ashcraft, W. R. Barbeck, Herman Blum, Jr., M. L. Brown, J. P. Dillard, C. Rollins Gardner, E. T. Gessell, D. C. Kieffer, V. C. Kneese, G. A. Linskie, Oslin Nation, J. A. Ray, P. N. Vinther.

Publicity—J. P. Ashcraft, *Chairman;* D. L. Echols, E. J. Hatzenbuehler.

Reception—W. R. Barbeck, *Chairman;* S. E. Ammons, A. F. Avera, J. A. Bishop, M. W. Brown, H. M. Cunningham, E. E. Farrow, D. W. Harrington, T. M. Jones, R. C. Knight, B. W. Levy, C. H. McLeod, M. J. Murray, H. G. Walcott, K. L. Wilson.

Sessions—P. N. Vinther, *Chairman;* J. W. Lacy, D. C. Pfeiffer, R. A. Walker.

Special Events—Herman Blum, Jr., *Chairman;* R. M. Burgess, Fred Downs, R. A. Mayes, I. P. Newby.

Transportation—J. A. Ray, *Chairman;* C. F. Gilmore, M. O. Tessman, Jr., C. C. Young.

1378

No. 1378

BY-LAWS
of
THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

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ARTICLE I
Organization

Section 1. **Organization.** THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS is a membership corporation, organized and existing under and by virtue of the laws of the State of New York.

Section 2. **Object.** The object of the Society is to advance the arts and sciences of heating, ventilating, cooling and air conditioning, and the allied arts and sciences, for the benefit of the general public.

Section 3. **Members.** The rights and privileges of a member of the Society shall be personal to himself and shall not be delegated or transferred, except that each member entitled to vote may vote in person or by proxy as hereinafter provided.

Section 4. **Government.** The Society shall be governed by its Charter and these By-Laws, the Rules promulgated by the Council in harmony therewith, and any amendments to the foregoing.

ARTICLE II
Memberships

Section 1. **Grades.** The grades of membership in the Society shall be designated as follows: (*a*) Honorary Members, (*b*) Presidential Members, (*c*) Life Members, (*d*) Members, (*e*) Junior Members, (*f*) Associate Members, (*g*) Affiliates. and (*h*) Students.

Section 2. **Construction.** Wherever in these By-Laws the words "MEMBER" or "MEMBERS" appear in upper-case letters (*sic,* "MEMBER," "MEMBERS"), they shall be taken to be synonymous with the terms "Honorary Members," "Presidential Members," "Life Members," and "Members"; and wherever the words "member" or "members" appear in lower-case letters (*sic,* "member," "members") and the context

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Adopted at the 56th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Dallas, Tex., January 23, 1950. (Effective January 26, 1950.)

permits it, they shall be taken to be synonymous with all the grades of membership in the Society.

Section 3. ***Qualifications.***

(*a*) *Honorary Members.* Any notable person of preeminent professional distinction may be elected an Honorary Member at an Annual Meeting of the Society, upon written proposal setting forth a resumé of the candidate's qualifications and attainments, signed by ten (10) MEMBERS and by two-thirds of the members of the Council.

(*b*) *Presidential Members.* Upon the installation of his successor, the outgoing President of the Society shall become a Presidential Member.

(*c*) *Life Members.* A Member, an Associate Member, or an Affiliate, in good standing, who has attained the age of sixty-five (65) years and has paid dues in said grades for thirty (30) years shall become a Life Member.

(*d*) *Members.* Any engineer or teacher of engineering over thirty-two (32) years of age, having eight (8) years or more of experience of a character satisfactory to the Council in the sciences relating to the arts of heating, ventilating, cooling or air conditioning, or the allied arts and sciences, of which four (4) years or more shall have been in responsible charge of important engineering work or in the teaching of engineering, shall be eligible to become a Member. Architects, chemists, physicians, scientists, and other persons over thirty-two (32) years of age, deemed by the Council to be qualified by reason of special experience in the foregoing arts and sciences, shall also be eligible to become Members.

Graduation from an engineering school approved by the Engineers' Council for Professional Development, or from a foreign engineering school maintaining similar standards, shall be deemed equivalent to four (4) years of such experience; from other technical schools maintaining four (4) year courses, three (3) years; from a non-engineering college, two (2) years; and each successfully completed year in an engineering college shall be deemed equivalent to one (1) year of such experience.

(*e*) *Junior Members.* Any person over twenty (20) and under thirty-three (33) years of age, having graduated from a college or school of engineering approved by the Engineers' Council for Professional Development, or from a foreign college or school of engineering maintaining similar standards, or possessing five (5) years of experience satisfactory to the Council in the sciences relating to the arts of heating, ventilating, cooling or air conditioning, or the allied arts and sciences, shall be eligible to become a Junior Member. Each successfully completed year in such college or school shall be deemed equivalent to one (1) year of such experience.

(*f*) *Associate Members.* Associate Members of the Society may hereafter remain in said grade. No person shall be admitted to Associate Membership in the Society following the adoption of these By-Laws.

(*g*) *Affiliates.* Any person over twenty-five (25) years of age, not necessarily an engineer but deemed by the Council to be qualified to cooperate with heating, ventilating, cooling or air conditioning engineers in the advancement of professional knowledge, shall be eligible to become an Affiliate.

(*h*) *Students.* Any person regularly enrolled and pursuing an engineering curriculum accredited by the Engineer's Council for Professional Development in a college or school of engineering, or in a foreign college or school of engineering maintaining similar standards, shall be eligible for student membership.

Section 4. ***Privileges and Limitations.*** Honorary Members, Presidential Members and Life Members shall be entitled to all the rights and privileges of a MEMBER. Associate Members shall not be eligible to hold elective office in the Society. Junior Members, Affiliates and Students may participate in all the activities of the Society

but shall not be eligible to vote or hold elective office in the Society. When a Junior Member shall attain the age of thirty-two (32) years, he shall apply and submit proof of his qualifications and experience for Member grade; upon his failure to apply and qualify for Member grade, he shall, when attaining the age of thirty-three (33) years, be transferred to Affiliate grade. Student membership shall cease at the end of the calendar year in which the Student's enrollment as a student has been terminated. The Council may transfer a Student to Junior Member grade on proof of his qualifications therefor or to Affiliate grade.

Section 5. ***Applications.*** An applicant for membership in the Society, or for advancement to a higher grade, shall submit his written application on a form approved by the Council, setting forth his qualifications, experience and references, and such other pertinent data as the Committee on Admission and Advancement and the Council may require.

Section 6. ***Prerequisites.*** A two-thirds affirmative vote of the Council shall be required for the admission of an applicant to membership or advancement to a higher grade. No applicant shall be admitted to membership or advanced to a higher grade unless his name shall have been published in an issue of the JOURNAL or mailed to all members.

Section 7. ***Resignation.*** Any member may resign his membership by writing to the Executive Secretary, who shall present the resignation to the Council at its next meeting. If the dues of the resigning member (pro-rated on a quarter-annual basis) shall have been paid to the date of the receipt of the resignation, his resignation shall be accepted.

Section 8. ***Suspension and Expulsion.***

(a) Non-Payment of Dues: If any member shall fail to pay his dues by April 1st, he shall be classed as delinquent and not entitled to vote; if such dues are not paid by July 1st, he shall be classed as not in good standing and his membership shall be suspended; if such dues are not paid by December 1st, the Executive Secretary shall notify the suspended member by registered mail that unless such dues are paid by December 31st, he shall cease to be a member of the Society, and upon his failure to cure such default by December 31st, his membership in the Society shall cease; provided that upon written application satisfactorily explaining a default, accompanied by payment of dues, the Council may, in its discretion, rescind any forfeiture of membership.

(b) Misconduct: The Council may, by a two-thirds vote of all the members thereof, censure, suspend or expel any member for misconduct in his relations to the Society, after written preferment of charges, thirty (30) days' written notice of hearing sent by registered mail, and an adequate opportunity to be heard before the Council or a committee of one or more MEMBERS designated by the Council.

Section 9. ***Effect of Termination of Membership and Lapse of Time.*** *(a)* All membership rights and interest in the property of the Society of persons resigning or otherwise ceasing to be members, or during the period of suspension of a suspended member, or on death, shall vest in the Society. *(b)* All acts of the Council which shall have received the express or implied sanction of the Society, shall be deemed to be the acts of the Society and shall not thereafter be impeached by any member. *(c)* No claim, demand, action or proceeding (individual, representative or derivative) shall be asserted or instituted by a member or former member against the Society, or any incumbent or former officer, Council or committee member thereof, by reason of service in any such capacities, after the expiration of six (6) months from the time when the same has accrued or actual or constructive knowledge has been acquired, or become available, of the facts upon which the same depends.

ARTICLE III

Chapters, Student and Special Branches

Section 1. ***Chapters.*** The Council may establish Chapters, which shall operate under the provisions of the Charter and the By-Laws of the Society, and the Rules and Regulations of the Council.

Section 2. ***Chapter Membership.*** Chapters shall be composed of at least twenty (20) members of any and all grades, exclusive of Chapter Student members, residing in the vicinity of the Chapter's headquarters, and only members of the Society in good standing shall be eligible to become and remain Chapter members. Chapter members shall hold the same grade of membership in the Chapter as are held by them in the Society. No member shall vote or hold office concurrently in more than one (1) Chapter of the Society. All grades of Chapter members, except Students shall be eligible to vote and hold office in Chapters.

Section 3. ***Limitations.*** The elected officers of Chapters shall receive no salary, emolument or compensation for their services as such. Chapters shall not act for the Society or subject the Society to any financial or other obligation, except such as the Society or the Council may by resolution assume. Notice to the foregoing effect shall be imprinted on the stationery used by each of the Chapters. Each Chapter shall promptly file a copy of its minutes with the Executive Secretary of the Society and make report to said Secretary of all of its proceedings. No contributions, except dues, shall be received or solicited by Chapters without the written approval of the Council. Chapters shall not issue publications or use the Society's name or emblem or Chapter insignia, without the approval of the Council. Chapters shall give no recommendations, endorsements or approvals of any scientific, literary, mechanical or engineering product for the promotion of private interests.

Section 4. ***Special Branches.*** Special Branches of the Society may be established, operated and maintained under the direction and in the discretion of the Council.

Section 5. ***Student Branches.*** Student Branches of the Society may be established, operated and maintained under the direction and in the discretion of the Council.

Section 6. ***Revocation of Charters.*** The charters of Chapters, Special Branches and Student Branches may be revoked by a two-thirds vote of all the members of the Council after written preferment of charges, sixty (60) days written notice of hearing sent by registered mail to the President of the Chapter or Branch, and an adequate opportunity to be heard before the Council or a committee of three (3) or more MEMBERS designated by the Council.

ARTICLE IV

Funds

Section 1. ***Admission Fees.*** Honorary Members and Students shall not be required to pay admission fees. The admission fees of Members, Junior Members and Affiliates shall be fixed by the Council, and shall be published in the JOURNAL. Admission fees shall accompany all applications for membership, and shall be refunded only in the event of the rejection of an application.

Section 2. ***Society Reserve Fund.*** Admission fees and such other funds as may from time to time be recommended by the Finance Committee and allocated by the Council,

shall be set aside and the principal thereof maintained as a Society Reserve Fund. Unless increased by the Society at an Annual or Special Meeting, the Society Reserve Fund shall be limited to a sum equal to fifteen dollars ($15.00) per member at the close of each fiscal year. The Council is hereby authorized and empowered in any fiscal year in which the Society's revenues may be insufficient to meet expenses, to utilize a maximum of twenty percent (20%) of the Society Reserve Fund as valued on the first day of the fiscal year in which such a withdrawal may be required.

Section 3. ***Dues.*** Honorary Members, Presidential Members and Life Members shall be exempt from the payment of dues.

Unless changed by the Society at an Annual or Special Meeting, the annual dues of Members, Associate Members and Affiliates shall be twenty-five dollars ($25.00) and of Students, three dollars ($3.00). The annual dues of Junior Members shall remain at ten dollars ($10.00) until January 1, 1951, whereupon the annual dues of Junior Members shall be fifteen dollars ($15.00).

Annual dues shall become due and payable in United States currency, or its equivalent, in advance on January 1st of each year, and shall in no case be subject to refund. Dues of new and advanced members, except Students, shall be pro-rated on a quarter-annual basis, and shall be payable on the first day of the month following notification of election or advancement, and if not paid within three (3) calendar months after such notification, the election or advancement shall be automatically rescinded. The Council in its discretion may suspend payment of dues by members in the Armed Forces in time of war, or may in its discretion defer or remit the dues of any member for special cause. Future annual dues may be compounded at a three percent (3%) rate by payment to the Society of the worth of an annuity equal to the member's dues for the period for which dues would be required. Compounded payments shall in no event be subject to refund.

Rule 1. The payment of Students' dues is hereby deferred until October 1st of each year. If such dues are not paid by December 31st, the delinquent Student's membership shall cease, and the Executive Secretary shall notify such Student by registered mail that his membership in the Society has ceased. Students' dues shall henceforth be applicable to the annual period commencing October 1st and terminating September 30th. As of October 1st, 1950, and on October 1st of each year thereafter, the Executive Secretary shall bill all students for dues in accordance with this Rule. ***(Adopted 10-22-50)***

Section 4. ***Dues to Constitute Subscriptions.*** Of the annual dues paid by members, a sum equal to the current subscription price shall be deemed to be a subscription for the JOURNAL. All members, except those whose annual or pro-rated dues shall amount to less than five dollars and fifty cents ($5.50), shall be entitled to receive the Society's periodical publications. Honorary Members, Presidential Members and Life Members shall be entitled to receive such publications without charge. Students shall be entitled to receive the JOURNAL without charge.

Rule 1. The Executive Secretary shall have imprinted on the Society's bills for dues payable by all members including Students, the following legend: "Includes $2.00 for one year subscription for the monthly JOURNAL of the Society, HEATING, PIPING AND AIR CONDITIONING." ***(Adopted 6-18-50)***

Section 5. ***Allocation of Dues for Research.*** Unless changed by the Society at an Annual or Special Meeting, forty percent (40%) of the dues received from Members, Associate Members, and Affiliates, shall be allocated for basic or fundamental research in the principles and laws underlying matters in the sciences relating to the arts of heating, ventilating, cooling and air conditioning, and the allied arts and sciences.

Section 6. ***Investment of Funds.*** *(a)* The Society Reserve Fund and such other funds as may be allocated by the Council for investment, shall be invested and reinvested in bonds and obligations of the United States Government, the Government of the Dominion of Canada, or in investments legal for trust funds under the laws of the State of New York, subject to the proviso that not less than one-half of such Fund

and funds shall be invested in bonds and obligations of the United States Government; (*b*) gifts and bequests to the Society for a specific purpose or purposes shall, after acceptance by the Council, be used for the purposes specified and invested in the manner directed by the donor or testator, or in the absence of such direction, in the manner provided in subdivision (*a*) hereof; (*c*) the Council is authorized and empowered, in its discretion and without liability to the Society or to any member thereof, to retain any gift or bequest in property (real, personal or mixed) in the manner and form in which it shall be at the time of such gift or bequest.

Section 7. ***Budget.*** The Finance Committee shall submit to the Council at the Council's last quarter-annual meeting, a Budget of estimated income and expenditures of the Society and all the committees thereof, for the succeeding fiscal year commencing November 1st and ending October 31st. The expenditures of the Society's funds shall be governed by the Budget as approved, modified, or from time to time amended by the Council, and no additional expenditures shall be made without the approval of the Council.

Section 8. ***Receipts and Disbursements.*** The Executive Secretary shall render all bills and collect all moneys due the Society, and shall enter all receipts in the Society's books and deposit the same to the Treasurer's Account in banks designated by the Council. Except as hereinafter provided, no contract or other obligation for the payment of money shall be valid unless signed or countersigned by the Executive Secretary. Except as hereinafter provided, all vouchers against the Society for the payment of funds shall be submitted to the Executive Secretary, who shall certify the correctness thereof and the authorization thereof by the Budget, and payment shall be made only upon such certification.

Section 9. ***Bonds.*** The Treasurer, the Executive Secretary, and all other officers, agents or employes authorized by the Council to endorse or execute drafts for the payment of money, shall give bond in a penal sum and with sureties approved by the Council, for the faithful performance of their duties, the premiums therefor, if any, to be paid by the Society.

Section 10. ***Audits.*** Between the close of the fiscal year and January 1st of each year, the accounts of the Society shall be audited by a certified public accountant approved by the Council, and the auditor's report shall be presented by the Treasurer at the Annual Meeting of the Society, and shall be published in the JOURNAL.

ARTICLE V

The Council

Section 1. ***Members.*** The Council shall consist of the last living Past President, the President, the First Vice President, the Second Vice President, the Treasurer and twelve (12) MEMBERS. The twelve (12) MEMBERS shall be divided into three (3) classes of four (4) in each class, and the MEMBERS in each class shall hold office for three (3) years and until their successors shall have been elected and installed. Four (4) MEMBERS shall be elected to the Council at each Annual Meeting, and also such additional number, if any, as may be necessary to fill vacancies. Pending the Annual Meeting, vacancies occurring in the Council may be filled by the Council. The Executive Secretary of the Society shall be the Secretary of the Council. He may take part in the deliberations of the Council but shall not be a member thereof or have any vote therein.

Section 2. ***Powers.*** In addition to the powers specifically conferred upon the Council (directors) by the laws of the State of New York, the Charter and these By-Laws, the Council shall have the general management and full control of the affairs and all activities of the Society and, subject to the Charter, the By-Laws and the laws of the State of New York, may, in its discretion, promulgate and amend Rules and Regulations, and issue directives for the administration of the Society's affairs and the regulation of all Committees, Chapters, Branches, Officers, Agents and Employes. The Council may, in its discretion, refer to the Society any important question pertaining to the Society, and shall refer any such question to the Society upon a majority vote taken at a stated or Special Meeting held by the Society.

Section 3. ***Meeting, Quorum and Reports.*** The Council shall hold its annual meeting as soon as practicable after the close of the Annual Meeting of the Society, and shall hold meetings quarter-annually thereafter. Special Meetings may be called by the President or by three (3) Council members. A majority of the Council members in office shall constitute a quorum. The Council shall keep a record of its proceedings, and shall report on its activities at each meeting of the Society and, at the Annual Meeting, it shall present a written report as required by the Membership Corporations Law of the State of New York.

Section 4. ***Notice of Council Meetings.*** Unless waived in writing or by telegraph or cable, notice of any quarterly or special meeting of the Council shall be given in writing, mailed to the last known address of each Council member, by the Executive Secretary or the President, or the three (3) Council members calling the meeting, not less than fifteen (15) nor more than thirty (30) days before the date fixed for the meeting.

ARTICLE VI

Officers

Section 1. ***Elected Officers.*** The elected officers of the Society shall be a President, a First Vice President, a Second Vice President and a Treasurer. The elected officers shall receive no salary, emolument or compensation for services rendered to the Society, and shall serve for one (1) year and until their respective successors shall be elected and installed.

Section 2. ***Appointed Officers.*** The Executive Secretary shall be appointed by the Council at its Annual Meeting, at a salary fixed by the Council, to serve for one (1) year and until his successor shall be appointed. The Executive Secretary shall be subject to removal by a two-thirds vote of the Council cast by secret ballot.

Section 3. ***Presiding Officer.*** At all meetings of the Society and of the Council, the President, or in his absence, the Vice Presidents in order of seniority, or in their absence the Treasurer or a MEMBER selected by the Council, shall preside.

Section 4. ***The President.*** The President shall exercise the powers and duties assigned to him by these By-Laws and, subject to the direction of the Council, he shall be the chief executive officer of the Society and generally supervise its affairs. At the Annual Meeting of the Society he shall make a report relative to the Society's condition, activities and progress. No President may be re-elected to that office until the Annual Meeting next following the expiration of his term.

Section 5. ***Vice Presidents.*** In the absence, disability, resignation or death of the President, the Vice Presidents in order of seniority, shall exercise the powers and perform the duties of the President.

Section 6. ***The Treasurer.*** The Treasurer shall have custody of the funds of the Society and the Society's books of account, which shall be open to the inspection of any member of the Council. Pending the Annual Meeting, a vacancy occurring in the office of Treasurer shall be filled by the Council.

Section 7. ***The Executive Secretary.*** The Executive Secretary shall be the chief administrative officer of the Society and the manager of the Society's publications. Copies of the minutes of all meetings of the Society, of the Council, and of all committees shall be filed with him. He shall keep such books, papers and records as the Society or the Council may direct, which shall be open to the inspection of any member of the Society. He shall be the keeper of the seal of the Society, and may in his discretion, use the designation "Secretary" on legal documents. He shall conduct the correspondence of the Society and shall keep full records of the same. He shall promptly notify the members of the Council, the officers, the nominated candidates, the members of all committees, and applicants for admission or advancement, of their election, nomination, appointment or advancement. He shall issue notices of all meetings of the Society, and, in the case of Special Meetings, he shall add a brief note of the object of the call. He shall be in charge of the offices of the Society and shall administer them under such rules of procedure as the Council may approve. Subject to the discretion of the Council, the Executive Secretary may employ such assistant secretaries and other personnel as may be deemed to be necessary.

Section 8. ***Other Duties.*** All officers of the Society shall perform the duties customarily attaching to their respective offices under the laws of the State of New York, and such other duties and services incident to their respective offices as are delegated to them in these By-Laws and as may from time to time be assigned to them by the Council.

ARTICLE VII

Committees

Section 1. ***Advisory Board.*** The Advisory Board shall consist of all Presidential Members, of which the last living Past President shall be chairman, and the said committee shall consider and make recommendations to the Council concerning matters of policy affecting the Society referred to the Board by the Council.

Section 2. ***Council Committees.*** Council Committees and the respective chairmen thereof, shall be appointed by the President, with the approval of the Council, as soon as practicable after the close of the Annual Meeting. Each Council Committee shall consist of three (3) Council members, who shall serve for a term of one (1) year and until their successors are chosen. The following shall be the Council Committees and their respective duties:

(a) Executive Committee, which shall investigate and make reports and recommendations to the Council regarding matters relating to the Society or any member or members thereof. During intervals between Council meetings the Executive Committee may exercise such powers of the Council as may lawfully be delegated to it by the Council.

(b) Finance Committee, which under the direction of the Council shall supervise and control the financial affairs of the Society and its books of account. It shall survey, investigate and analyze all financial requirements and expenditures, scrutinize all Budget estimates, and prepare the Budget for submission to the Council. The Treasurer shall be an ex-officio member of the said committee, with the power to vote.

(c) Membership Committee, which shall publicize the aims, activities, achievements and the scientific and educational purposes of the Society, toward the end that persons duly qualified shall apply for membership therein.

(d) Program and Papers Committee, which shall plan the general character of all technical meetings of the Society, solicit and receive papers for consideration by the Publication Committee, and select from the papers which have been approved by the Publication Committee those for presentation at technical sessions.

(e) Standards Committee, which shall consider all scientific questions and data pertaining to engineering codes and standards, initiate and propose changes and improvements thereof for the public benefit and report its recommendations to the Council.

Section 3. ***General Committees.*** Unless otherwise provided, the General Committees, and the respective Chairmen thereof, shall be appointed by the President, with the approval of the Council, as soon as practicable after the close of the Annual Meeting. The President, with the approval of the Council, shall at the same time appoint the Chairman and the Vice Chairman of the Committee on Research. The Chairman of said Committee shall have had at least one (1) year of service on said Committee. The Vice Chairman shall perform the duties of the Chairman in the latter's absence, disability, resignation or death. The following shall be the General Committees and their respective duties:

(a) Admission and Advancement Committee, consisting of three (3) MEMBERS in good standing for at least ten (10) years and having at least three (3) years of service on the Council or on General Committees, which shall receive and consider applications for membership and advancement, make diligent scrutiny and inquiry as to the character and qualifications of applicants, and report to the Council on the eligibility of each applicant for admission or advancement. The correspondence and proceedings of said committee shall be secret and confidential and its correspondence concerning unsuccessful applicants shall be destroyed within a reasonable time.

(b) Publication Committee, consisting of three (3) MEMBERS. Subject to the direction of the Council the said committee shall formulate the editorial policies of the Society and for all of its publications. The Chairman of the said Committee may appoint sub-committees of one (1) or more members to review and report to the Committee on the quality and appropriateness for publication of papers and bulletins intended for presentation or presented at Society meetings and the discussions thereof. In the performance of its functions the said Committee and its sub-committees shall be subject to the following conditions: *(a)* That the data recommended for publication shall tend toward the professional education of the individual engineer; *(b)* that such data shall be free from commercial bias; and *(c)* that such data shall tend to advance for the public benefit, the sciences relating to the arts of heating, ventilating, cooling or air conditioning, or the allied arts and sciences.

(c) Guide Committee, consisting of nine (9) MEMBERS. The said committee, in harmony with the editorial policies of the Society, shall compile the text section of THE GUIDE. The Director of Research shall be an ex-officio member of the said committee.

(d) Charter and By-Laws Committee, consisting of three (3) MEMBERS, which shall consider all matters affecting the Charter and By-Laws, Rules and Regulations, and make recommendations thereon to the Council.

(e) F. Paul Anderson Committee, consisting of the First Vice President and four (4) MEMBERS, to serve for a term of one (1) year, which shall receive from the Council the announced conditions for the making of the subsequent annual award of the F. Paul Anderson Medal, select, solicit and carefully consider candidates for the award, and make nominations to the Council of the members complying with the announced conditions and deemed worthy to be recipients of the award. The decision to confer the award and all matters relating thereto shall be by a majority vote of all the members of the Council and in accordance with the presentation letter of Thornton Lewis.

(f) Chapter Relations Committee, consisting of a member of the Council, who shall serve for a term of one (1) year, and six (6) MEMBERS. Each of the seven (7) regional areas, as set forth in **Appendix A** hereof, shall have representation on said Committee. The said Committee shall investigate applications for the creation of Chapters and Student and other Branches, report thereon to the Council and, subject to the direction of the Council, assist in the organization or reorganization thereof. The said Committee shall review the organization and activities of Chapters and Branches, and make recommendations to the Council on policies and procedures pertaining to their operations, activities, programs, speakers and meetings.

(g) Chapters Conference Committee, consisting of one (1) member and one (1) alternate member selected by each Chapter, to serve for a term of one (1) year from the opening of the Annual Meeting of the Society. The said committee shall choose one (1) of its members as its chairman. It shall elect the MEMBERS and the alternate to serve on the Nominating Committee and promptly notify the Executive Secretary of such election. It shall consider matters affecting the operations of the Chapters, and make annual reports thereon to the Council. Each alternate member may be present at committee meetings but shall not participate in the deliberations thereof or vote therein, except in the absence, disability, resignation or death of the committee member for whom he is alternate. The chairman of the Chapter Relations Committee shall be an ex-officio member of this committee.

Rule 1. The Chairman of the Retiring Chapters Conference Committee and other such officers as said Committee may have chosen, shall continue in their respective offices on the succeeding committee until their successors are chosen. *(Adopted 10-22-50)*

(h) Committee on Research, consisting of fifteen (15) MEMBERS, nominated by the Council or as provided in **ARTICLE VIII**, *Section 4,* and elected by the Society in the manner of elected officers. Subject to the direction of the Council, the said committee shall conduct and coordinate fundamental and basic research, and study and determine the principles and laws underlying matters in the sciences relating to the arts of heating, ventilating, cooling and air conditioning, and the allied arts and sciences, and cooperate with universities, colleges, schools and other organizations and groups, including governmental agencies, in the investigation of research subjects, subject to the proviso that all of the foregoing activities shall be devoted to the public welfare and general benefit, and shall not be designed to promote any individual, private or commercial interests. No contributor shall be specially favored on account of any contribution for research, which shall be used only for the public welfare. The said committee shall submit to the Publication Committee and to the Executive Secretary of the Society, its papers and reports concerning its investigations and activities.

There shall be a *Research Executive Committee,* consisting of the chairman, the vice chairman and three (3) other members of the Committee on Research appointed by the chairman, which shall exercise the powers and carry out the purpose of the Committee on Research during intervals between Committee meetings.

The chairman of the Committee on Research shall appoint such *Technical Advisory Committees* as may be deemed expedient, to advise the Committee on Research and the Director of Research on specific research projects. At least one (1) member of each Technical Advisory Committee shall be a member of the Committee on Research, and the chairman of the Committee on Research and the Director of Research shall be ex-officio members thereof. Technical Advisory Committees shall be governed by such rules and regulations as may be recommended by the Committee on Research and adopted by the Council.

The chairman of the Committee on Research may appoint a *Technical Adviser* as the Committee's consultant, who shall serve without compensation.

The Committee on Research shall, as soon as may be practicable after the close of each Annual Meeting of the Society, recommend to the Council for appointment a *Director of Research,* whose activities, proceedings and reports shall be subject to the direction of the committee and the approval of the Council. The salary

of the Director of Research shall be fixed by the Council upon the recommendation of the Committee on Research, and his employment may be terminated in the Council's discretion.

The Director of Research shall direct the operations of the Society's research program. Subject to the approval of the Committee on Research, the Director of Research may employ such assistants and other personnel as may be deemed by said Committee to be necessary. The Director of Research shall be provided with a revolving fund in an amount fixed by the Council upon the recommendation of the Finance Committee, for the payment of the compensation of part time or temporary employees, travelling expenses and incidental petty cash outlays. He shall approve all purchase invoices and disbursements for research, and submit vouchers for the payment thereof at least semi-monthly to the Executive Secretary. The Director of Research shall assist the Committee on Research in the preparation of its Budget for submission to the Finance Committee of the Society. The Director of Research and such other research assistants as may be employed in connection with research, shall, in consideration of such employment, agree in writing that any inventions, discoveries, ideas, plans, processes, formulae, experimental results and information received, published, divulged, made, or developed, or which may be in the course thereof while engaged in such employment, shall belong to the Society, and that no material relating thereto will be submitted elsewhere without the Council's consent.

Section 4. ***Other Committees.*** The Council may from time to time appoint other committees of one (1) or more members, and define their powers and duties, and it may abolish any such committees.

Section 5. ***General Provisions Concerning Committees.*** All Council, General and other committees of the Society, except the Nominating Committee, are subject to the following provisions:

(a) Except as otherwise provided, members on General Committees shall be divided into three (3) classes of equal number, each class to serve for three (3) years, and one-third of the prescribed number to be appointed, or in the case of the Committee on Research elected, annually.

(b) Except as otherwise provided, the committee year of all committees shall run from the close of the Annual Meeting to the close of the next Annual Meeting. The committee members whose terms expire shall continue in office until their respective successors are appointed or elected.

(c) Each committee shall meet as soon as practicable after its appointment or election, and may hold semi-annual meetings and such special meetings as the chairman or one-third of the members thereof may call. Each committee may recommend rules of procedure, which shall become operative upon the approval of the Council. Unless waived in writing or by telegraph or cable, notice of all semi-annual and special meetings of all committees shall be given in writing to all committee members and to the Executive Secretary not less than fifteen (15) nor more than thirty (30) days before the date fixed for the meeting. At committee meetings a majority of the members in office shall constitute a quorum. The committees shall promptly and fully report their activities to the Council and file with the Executive Secretary the minutes of their meetings and a complete record of their proceedings.

(d) The Council may by a two-thirds vote, remove a member of any committee, and may, by a majority vote, designate a member to fill the vacancy or any other vacancy arising.

(e) Each committee shall submit to the Finance Committee on or before September 15th of each year, an estimate of its disbursements for the ensuing fiscal year, and no committee, or member thereof, shall have the authority to incur any indebtedness or pecuniary obligation for which the Society shall be responsible, or claim reimbursement for advances, except to the extent authorized in the Budget or by resolution

of the Council. Each committee member shall be responsible for the proper application of all funds remitted to him.

(f) Each committee's actions, proceedings, findings, conclusions and reports shall be subject to the direction and review of the Council, and the Council may take such steps, or see that such steps are taken by the committees as may be appropriate to comply with the Charter and By-Laws, and to make effective any resolution adopted by the Society or any resolution, rule or directive of the Council.

(g) If any doubt or controversy should arise as to whether a particular subject or matter is within the jurisdiction of a committee or whether any action should be taken by a committee, or in the case of a committee tie vote, the same shall be settled and determined by the Council.

(h) Except as otherwise provided, ex-officio members of committees may participate in committee deliberations but shall have no vote therein.

ARTICLE VIII

Meetings, Nominations and Elections

Section 1. ***Meetings.*** The Annual Meeting of the Society shall commence during the thirty (30)-day period beginning with the fourth Monday in January, and shall continue from day to day as the Council may arrange. Semi-Annual Meetings shall be held at such times as may be fixed by the Council. Special Meetings may be called at any time by the Council, and shall be called by the Council upon the written request of the President or of fifty (50) MEMBERS of the Society. Meetings shall be held at such place or places as the Council may designate, and shall be governed by Robert's Rules of Order, Revised, except when inconsistent with the laws of the State of New York, the Charter or these By-Laws. At any meeting of the Society, the presence of fifty (50) members entitled to vote shall be necessary to constitute a quorum.

Section 2. ***Notices.*** Notice of the Annual, Semi-Annual and of any Special Meeting of the Society shall be given in writing by the Executive Secretary and mailed, postage prepaid, not less than twenty (20) nor more than forty (40) days before the date fixed for the meeting, to each member of the Society at his last known address appearing on the records of the Society, and shall be published in the JOURNAL. Notices of Special Meetings shall state the purpose or purposes for which the meeting is called, and no business other than that set forth in the notice shall be entertained or transacted thereat.

Section 3. ***Nominating Committee.*** The Nominating Committee shall consist of eleven (11) MEMBERS and two (2) alternate MEMBERS all of whom shall have been in good standing as MEMBERS for a period of at least five (5) years. Four (4) MEMBERS and one (1) alternate MEMBER of said Committee, each from a different regional area, as set forth in **Appendix A** hereof, shall be elected by the Council at its last quarter-annual meeting; and seven (7) MEMBERS, each from a different regional area, as set forth in said Appendix, and one (1) alternate MEMBER of said Committee shall be elected by the Chapters Conference Committee by the close of the second day of the Annual Meeting of the Society. Each alternate MEMBER may be present at committee meetings but shall not participate in the deliberations thereof or vote therein except in the absence, disability, resignation or death of a MEMBER of the group with which he was elected. No Council member shall be eligible to serve on the Nominating Committee. No Chapter shall

be represented on the Nominating Committee by more than one (1) member. The Nominating Committee so chosen shall effect its own organization during the Annual Meeting of the Society, and shall hold a meeting during the Semi-Annual Meeting of the Society, at which seven (7) MEMBERS entitled to vote on said Committee shall constitute a quorum. The round-trip railroad fare and lower berth expenses incurred by Committee members and alternates in attendance at the Semi-Annual Meeting, shall be included in the Budget and defrayed by the Society. By September 1st of each year, the Nominating Committee shall nominate candidates for the elective offices and members of the Council to be elected at the ensuing Annual Meeting, and shall notify the Executive Secretary of the names of the nominees, the notice to be accompanied by the written acceptances of the candidates.

Section 4. ***Other Nominations.*** Nominations of officers and members of the Council, other than those nominated by the Nominating Committee, and nominations of members of the Committee on Research, other than those nominated by the Council, may be made in writing by at least fifty (50) members eligible to vote, upon presentation of such nominations, with each nominee's consent, to the Executive Secretary at least sixty (60) days prior to the opening of the first session of the Annual Meeting, whereupon the nominees' names shall be placed upon the ballot with a notation that they are presented by members independent of the Nominating Committee.

Section 5. ***Voting.*** Voting at any meeting may be in person or by proxy, but only the Executive Secretary and MEMBERS of the Society shall be eligible to act as proxies. Proxies shall not be valid for more than three (3) months from dates of execution. The Executive Secretary and the MEMBERS acting as proxies shall hold the ballots of their principals secret and confidential. Voting for election of officers, Council members, members of the Committee on Research, on proposals to amend these By-Laws, and on questions required to be referred to the Society pursuant to **ARTICLE V,** *Section 2,* shall be by secret ballot. In the event of any tie vote, the Council shall decide the vote.

Section 6. ***Ballots.*** Together with notice of the Annual Meeting, the Executive Secretary shall forward appropriate proxies and ballots to members entitled to vote. The proxies and ballots shall contain spaces for write-in names.

Section 7. ***Results.*** The polls for election shall be opened at the opening of the Annual Meeting and shall remain open for a period of five (5) hours. Thereafter the ballots shall be opened by three (3) inspectors of election appointed by the President, who shall be authorized to fill any vacancy occurring among such inspectors. The inspectors of election shall consider ballots and votes to be valid provided the intent of the voter is clear. The result of the vote shall be reported by the inspectors of election in writing, and shall be announced by the President on the second day of the Annual Meeting, whereupon the terms of the inspectors of election shall expire. The elected candidates shall be installed, and their terms shall commence, at the close of the last session of the Annual Meeting.

ARTICLE IX

Amendments

Section 1. ***Prerequisites.*** These By-Laws may be amended by a two-thirds vote of the Society at an Annual Meeting thereof, provided that written notice of the proposed amendment, subscribed by two-thirds of the members of the Council or by

fifty (50) MEMBERS, be given at a previous stated or Special Meeting, and that notice thereof as pertinently amended by majority vote at said stated or Special Meeting, be also given by the Executive Secretary in the notice of the Annual Meeting.

Section 2. ***Renumbering.*** The Council may by a two-thirds vote, renumber existing Articles or Sections of these By-Laws.

ARTICLE X

Adoption

Section 1. ***Effect.*** These By-Laws shall supersede all previous Constitutions, By-Laws and Rules of the Society, and shall come into effect upon the adjournment of the meeting at which these By-Laws shall be adopted.

BY-LAWS—APPENDIX A

Regional Areas

Region 1.

- Maine
- Vermont
- Massachusetts
- New Hampshire
- New York
- Rhode Island
- Connecticut

Region 2.

- Ohio
- New Jersey
- Pennsylvania
- Delaware
- Maryland
- Dist. of Columbia

Region 3.

- Indiana
- Illinois
- Michigan
- Wisconsin

Region 4.

- Minnesota
- Iowa
- Nebraska
- Missouri
- Kansas
- North Dakota
- South Dakota
- Montana
- Wyoming
- Colorado
- New Mexico

Region 5.

- Idaho
- Utah
- Nevada
- Washington
- Oregon
- Arizona
- California

Region 6.

- Virginia
- West Virginia
- North Carolina
- South Carolina
- Florida
- Georgia
- Alabama
- Mississippi
- Kentucky
- Tennessee
- Louisiana
- Arkansas
- Oklahoma
- Texas

Region 7.

The Provinces of Canada.

1379

No. 1379

NIGHT-AIR COOLING

By F. E. Giesecke,* New Braunfels, Tex.

In localities where the night-air temperature is considerably lower than the day-air temperature, it may be economical to use night-air for cooling.

For the San Antonio, Tex., area, records of the U.S. Weather Bureau give the following data (Fig. 1):

	Avg Minimum Daily Temp F deg	Avg Maximum Daily Temp F deg
July 1948	75.2	93.6
Sept. 1948	67.9	86.6

Assuming that in buildings in the San Antonio, Tex., area which are air conditioned during the day, the maximum night-air temperature will be 80 F, it will be possible to exhaust 80 F air and replace it with 75 F air in July and with 68 F air in September and thereby effect considerable cooling in the San Antonio, Tex., area. In buildings which are not air conditioned, the maximum night-air temperature will be considerably higher than 80 F and the benefits that can be secured with night-air cooling will be correspondingly greater than in air conditioned buildings.

For a number of years, the author has used night-air cooling in his residence by leaving the windows open during the night, so that the contents of the building and the building structure could cool, and then keeping the windows and blinds closed during the day so as to exclude heat as much as possible. As a result, the indoor air temperature during the afternoon was from 10 to 15 deg lower than the corresponding outdoor air temperature. This degree of cooling produces fairly comfortable conditions within the building at practically no cost.

During the summer of 1949 the author had the opportunity of using night-air cooling in the building (Fig. 2) of the First National Bank of New Braunfels, Tex. The banking room is 44×46 ft and 20 ft high; its volume is about 40,500 cu ft and the surface area of walls, floor, and ceiling is about 7600 sq ft.

In 1931, at the time of its construction, the building was equipped with a forced warm-air heating system, which included a centrifugal fan belted to a

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* Consulting Engineer and past president and Life Member of A.S.H.V.E.

Presented at the 56th Annual Meeting of The American Society of Heating and Ventilating Engineers, Dallas, Texas, January 1950.

5 hp, 1800 rpm, 220 volt motor so as to drive the fan at 648 rpm, and to deliver air at the rate of about 9000 cfm. The air was delivered to the banking room through ceiling outlets.

During the spring of 1949 the bank building was equipped with a 15 hp air conditioning system but the 5 hp fan for the heating system was left in the building. The duct connections were changed so that the fan could draw the warm air out of the banking room through the ceiling openings and discharge it outside of the building through the duct, which is used as the fresh-air inlet during the day. As the warm air was drawn out of the room it was replaced by cool outdoor air entering through windows opened for that purpose.

Since the officials of the bank desired to maintain the air temperature at about 78 F during working hours, it was decided, after some experimenting, to

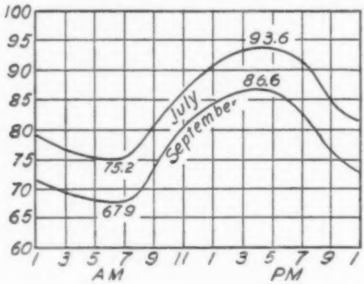

FIG. 1. AVERAGE DAILY TEMPERATURES DURING JULY AND SEPTEMBER, 1948, AS RECORDED BY THE U.S. WEATHER BUREAU AT SAN ANTONIO, TEX.

operate the air conditioning system from about 7 a.m. to about 5 p.m. and the night-air cooling system from about 9:30 p.m. to about 6:30 a.m.

REDUCTION OF TEMPERATURE BY NIGHT-AIR COOLING

The resulting temperatures for the week beginning September 26, 1949, are shown in Fig. 3. It will be seen that, on Tuesday, September 27, the temperature was about 77 F during the working hours. When the air conditioning unit was turned off (at about 5:00 p.m.) the temperature rose to about 79 F and then, when the night-air cooling system was turned on (at about 9:30 p.m.) the temperature gradually fell to 77 F and was then maintained at that temperature by the air conditioning unit during the working hours of Wednesday. After that, the same order of operation was repeated.

The temperatures shown in Fig. 3 are those of the air about 6 in. above the floor because the recording thermometer was placed on the floor. Near the room ceiling the air temperature was no doubt a few degrees higher.

It may seem strange that the indoor air temperature should rise 2 deg from 5:00 p.m. to 10:00 p.m. when there was no heat producing appliance in the building. The explanation is quite simple. When a room is air conditioned, the air in the room is cooled and the cool air, in turn, cools the enclosing walls, floor, and ceiling, as well as the persons, furniture, etc. in the room. It follows that, so long as the walls, etc., are being cooled by the air, the walls are warmer than the air and that as soon as the air conditioning system ceases to function, the air begins to be heated by the walls, ceiling, and floor. For example, if the

surface temperature of the walls, floor, and ceiling is 2 deg higher than the temperature of the room air and if the film coefficient is 1.65 and the surface area of the walls, floor, and ceiling is, together 7600 sq ft, heat will be delivered to the room by those surfaces at the rate of about 25,000 Btu per hr.

To determine the amount and the cost of the night-air cooling it was found that the fan draws 17 amp of 220 volt current and removes warm air at the rate of about 6000 cfm.

The difference between the 6000 cfm moved and the 9000 cfm originally contemplated is probably the result of having more friction in the long and complicated duct system than the estimated friction.

FIG. 2. FIRST NATIONAL BANK BUILDING IN NEW BRAUNFELS, TEX.

COOLING COSTS

Electric current costs the bank about 2 cents per kwhr, hence the cost of moving 6000 cfm is $17 \times 220 \times 0.002$ or 7.48 cents per hour. If the difference between the temperature of the warm air drawn out of the building and that of the cool air replacing it is 10 deg, the quantity of heat removed from the building is

$$6000 \times 60 \times 0.075 \times 0.24 \times 10$$

or 64,800 Btu per hour. Since it costs 7.48 cents per hour to remove 64,800 Btu per hour, the current cost per ton is

$$7.48 \times 12{,}000/64{,}800,$$

or 1.39 cents per hour (if the current costs 2 cents per kwhr).

The difference between the temperature of the air drawn out of the building and the temperature of the air replacing it was determined by a two-pen recording thermometer. This was placed outside the building so that the instrument was about 6 ft above the ground and 2 ft from the building; the temperature bulb was about 6 in. above the ground and in front of the lower part of the louver through which the air was blown out of the building.

The records made by the two pens are shown in Fig. 4. In the record for Friday night, it appears that the fan came on at 8:00 p.m. and exhausted air at 76 F; also that the fan went off at 5:00 a.m. when the exhaust air had a temperature of 72 F. At the same periods the outdoor air temperatures were,

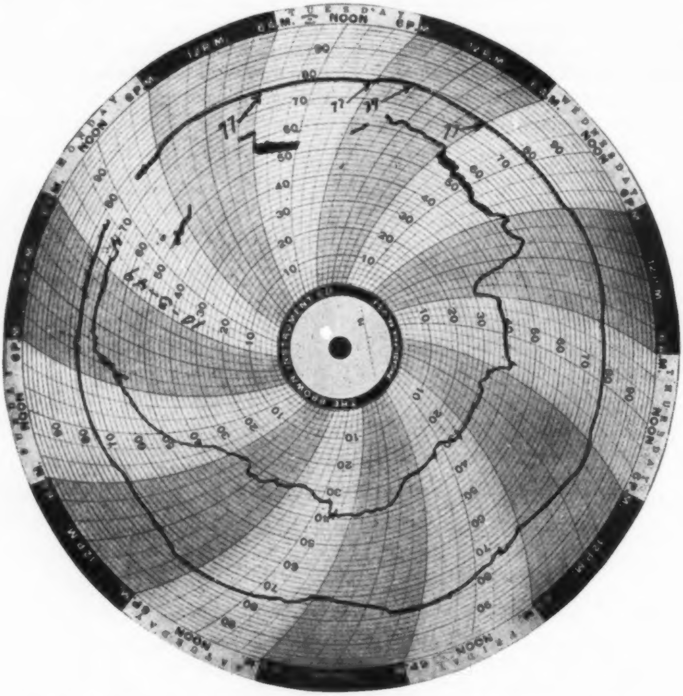

Fig. 3. Indoor Air Temperatures, at Floor Level, During the Week Beginning September 26, 1949.

respectively, 66 F and 54 F, so the average difference between the exhaust and outdoor air temperatures was 14 deg and the cost of cooling was at the rate of $1.39 \times 10/14$ or 0.99 cents per ton per hour during Friday night.

However, while the chart of Fig. 4 shows the temperature of the exhaust air to vary from 76 F to 72 F, the chart of Fig. 3 shows that the indoor air temperature near the floor was practically 80 F during every night and hence the air temperature at the ceiling must certainly have been at least 80 F.

An explanation of this difference between the two charts is that the exhaust air came out of the louver with such a velocity that it carried outdoor air with it and the temperature bulb measured the temperature of a mixture of exhaust air and outdoor air. At 8:00 p.m. about 71 percent of the mixture might be

exhaust air and 29 percent, outdoor air. At 5:00 a.m. about 69 percent of the mixture is exhaust air and 31 percent, outdoor air.

If it is assumed that the exhaust air had a temperature of 80 F instead of from 76 F to 72 F as shown in Fig. 4, the cost of cooling is reduced to 0.99 × 14/20, or about 0.70 cents per ton per hour or, for the 9-hour period of Friday night, about 63 cents for electric current.

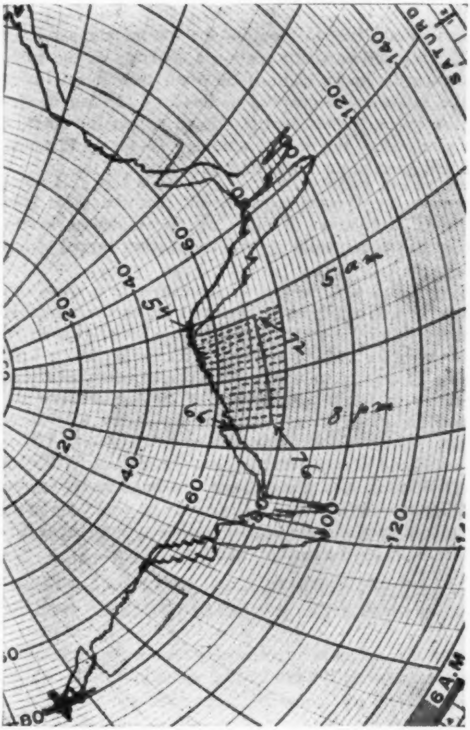

Fig. 4. Temperatures of Exhaust Air and Make-up During Night-Air Cooling

Conclusions

Granting that the calculations used in this paper are approximations and are based on a single set of observations, it has been shown that night-air cooling can be applied economically to commercial and industrial buildings as well as to residences.

According to the data presented here, if the cost of mechanical refrigeration in small units is about 1 kw per ton, the cost of night-air cooling should be less than one-half the cost of mechanical refrigeration.

Night-air cooling evidently cannot be used in localities where the outdoor air contains dust or other objectionable materials in such amounts that it would be detrimental to pass large quantities of air through the building.

DISCUSSION

Lester T. Avery, Cleveland. Ohio: Night-air cooling can be done to advantage. I do not accept the statement you cannot use it when you have atmospheric dust because we have methods of cleaning air which will take any impurities out.

Your difficulties with night-air cooling are where you have fog, high humidity, which give you a tendency for mildew and mold in the home. One thing that I brought up a year ago on the subject of home cooling has not been investigated but has been emphasized in this paper. It is the storage capacity of the building structure for moisture. Moisture storage during night-air cooling, where the moisture is

absorbed in the insulating materials and building materials and then re-evaporates in the daytime under the sun on the walls and roofs, gives you evaporative cooling.

Mr. Heisterkamp who is responsible for the thought insists that their tests indicate something of that kind is going on; and that is why when you use refrigeration, it requires from one-half to two-thirds of the figured capacity to do cooling because you take the advantage of the night by storing water vapor which gives you re-evaporation in the daytime.

J. S. HOPPER, Arlington, Tex.: I would like to mention the use of the attic fan here in Texas as an example of the utilization of the night-air cooling. In the early evening, the outside atmosphere is usually more comfortable than that found inside of the buildings. By using the attic fan, a large quantity of this cool air can be circulated through the house and bring the inside temperature down within the comfort region. The fans should not be used when the sun is shining. In fact, the house should be closed except for minimum ventilation requirements.

P. R. ACHENBACH, Washington, D. C.: I want to point out that the same kind of results can be obtained by gravity circulation of night air in residences in the Western states as are described for this bank building in Texas using forced circulation of cool night air.

This past summer I was observing indoor and outdoor temperatures in a two-story frame residence in Wyoming where it was the practice to open all the windows at night and close them in the morning, leaving most of the window shades drawn. In that locality on warm days the temperature usually reached a maximum about 1:00 p.m. and often decreased as much as 20 deg by sundown. Under such conditions with night cooling it was quite common to maintain a difference of 20 deg between the maximum outdoor temperature and maximum indoor temperature in the afternoon. I recall that on two particular days when the temperature reached 105 and 110 F outdoors, the maximum temperature observed indoors was on the order of 85 F.

While the heat storage characteristics of a house undoubtedly have an effect on the indoor-outdoor temperature difference during such hot periods of short duration, judicious manipulation of the windows and window shades makes a noticeable difference in the maximum indoor temperatures during hot weather.

AUTHOR'S CLOSURE: Mr. Avery's suggestion that night air can be cleaned is correct; however, in some cases, cleaning would not be justified by the benefits resulting from night-air cooling.

The suggestion that night air may also be used to store moisture in the insulating and other materials of a building seems very important and should be followed by further study.

Mr. Achenbach's description shows clearly that night-air cooling is fully as effective in Wyoming as it is in Texas.

Dean Hopper's suggestion that attic fans should not be used when the sun is shining should be modified because comfort conditions in rooms can be improved by drawing a stream of warm outdoor air through them even if the indoor temperature is raised thereby.

Dr. Ernst Schmidt, professor of thermodynamics at the University of Brunswick, has given considerable study to night-air cooling, especially for use in desert locations where electric current is not available for refrigeration, and has applied for patents in France and Germany on a system in which night air is used for storing cold in a mass of broken stone and the stored cold is used for cooling.

1380

No. 1380

PHYSIOLOGIC EXAMINATION OF THE EFFECTIVE TEMPERATURE INDEX

By NATHANIEL GLICKMAN*, TOHRU INOUYE**, ROBERT W. KEETON†, M.D., CHICAGO, ILL., AND MAURICE K. FAHNESTOCK††, URBANA, ILL.

This paper is the result of research sponsored by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and by the United States Public Health Service in cooperation with the University of Illinois, College of Medicine

IN RECENT years a number of investigators[1-4] have indicated that the effective temperature index places too much emphasis on the influence of relative humidity in determining sensations of warmth and coolness at temperatures customarily encountered in heating and ventilating practice and too little emphasis at the high temperatures. Houghten[5], endeavoring to explain the discrepancies, stated that most of the possible variations in the accuracy of the effective temperature lines fall within a range of from ½ to 3 deg effective temperature. Bedford[6], on the other hand, indicated that the index is probably very reliable at ordinary temperatures but that some adjustment is needed at the very high temperatures.

These reports were based upon experiments with subjects in approximate equilibrium with the environment. The subjects of Winslow et al[1] were exposed to different ambient environments for two or more hours; the subjects of Yaglou[3] for three hours; and the data analyzed by Rowley et al[2] were for subjects exposed for one to two hours. The effective temperature index, on the other hand, was developed from instantaneous thermal impressions of subjects passing back and forth between conditioned rooms. It is therefore conceivable that such an index developed under dynamic conditions would not be strictly applicable for equilibrium conditions.

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†† Research Professor of Mechanical Engineering and Engineering Director of the Physical Environment Unit, Department of Mechanical Engineering, University of Illinois. Member of A.S.H.V.E.

[1] Exponent numerals refer to References.

Presented at the 56th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Dallas, Tex., January 1950.

Previous contributions from this laboratory[7-9] have discussed certain physiological adjustments of human subjects on exposure to sudden change in environmental temperature and humidity. The purpose of the present study was to compare, by means of physiological measurements, the response of subjects to two different ambient environments having the same effective temperature. Relative humidities of 30 and 80 percent were selected for each effective temperature studied. Effective temperatures in the zones of body cooling, thermal neutrality and evaporative regulation were selected. The work was conducted with the subjects in approximate equilibrium with the environment as well as on exposure to sudden change in environment.

SYMBOLS AND ABBREVIATIONS

ET = effective temperature
DBT = dry bulb temperature
psi = pounds per square inch
RH = relative humidity
MT_S = mean skin temperature
T_R = rectal temperature
P = probability

TEST MEASUREMENTS

Subjects: Fifteen healthy young adult male students, ranging in age from 19 to 29 years (average 23), participated in the experiments. Table 1 presents the physical description of the subjects and the effective temperatures to which they were exposed. Seven subjects were studied at each effective temperature.

Procedure, Observations and Experimental Conditions: The procedure and observations have been described in the papers[7-9] cited. The subjects, dressed in thermocouple union suits, remained in room 1 (previously designated comfortable room 1) for one hour and then entered and remained in the hot room for one hour after which they returned to the first room now designated room 2 (previously designated comfortable room 2).

Observations included skin and rectal temperatures, evaporative weight loss, pulse rate, and thermal sensation.

The scale for thermal sensation was as follows: (1) Cold; (2) Cool; (3) Slightly Cool; (4) Comfortable; (5) Slightly warm; (6) Warm; (7) Hot. The subjects were trained in the use of the scale before starting the experiments. Intermediate votes such as 2½, 3½, or 4½ were accepted during the adjustment period.

The hot room was maintained, as in previous studies[7-9], at a dry bulb temperature (DBT) of 98.5 F $\pm$1.0 deg with water vapor pressure of 0.599 $\pm$0.036 pounds per square inch (psi)* corresponding to a relative humidity (RH) of 66 $\pm$4 percent and 90.2 ET.

Ten ambient conditions were used in room 1 for which the effective temperature, dry bulb temperature, relative humidity and water vapor pressure are presented in Table 2. Environment I represents the lowest effective temperature, II the next lowest, etc. with V representing the highest effective tempera-

* 1 psi = 51.715 mm Hg.

TABLE 1—DESCRIPTION OF EXPERIMENTAL SUBJECTS

SUBJECT	AGE YEARS	HEIGHT FT AND IN.	WEIGHT LB	AREA SQ FT	ET[a]
C.A.	29	5 7	173	20.5	68.8, 73.4
J.B.	24	5 11¾	137	19.5	68.8, 73.4
M.B.	20	5 8½	136	18.7	68.8, 73.4
T.B.	25	5 11	160	20.7	59.6, 68.8, 73.4, 78.0, 82.6
H.I.	21	5 6	125	17.9	78.0
B.K.	24	5 9	139	18.8	68.8, 73.4
E.K.	23	6 0	190	22.6	59.6, 68.8, 73.4, 78.0
H.K.	21	5 9½	138	19.0	82.6
N.L.	25	5 6	159	19.6	82.6
Ab.R.	19	5 9	141	18.8	59.6, 78.0, 82 6
Ad.R.	21	5 8½	191	21.2	59.6, 78.0, 82.6
G.S.	28	5 11	188	22.0	59.6
J.S.	20	6 0	135	19.5	68.8, 73.4
R.S.	21	6 1	168	21.1	59.6, 78.0, 82.6
N.Z.	21	6 0	152	20.4	59.6. 78.0. 82.6

[a] Environments to which the subjects were exposed (two experiments at each effective temperature).

ture. All environments designated by an A are those with 80 percent RH and all by a B with 30 percent RH. For example, environment II A would have the same effective temperature as II B, but a higher relative humidity and lower dry bulb temperature.

Statistical Analysis of the Data: Student's[10-11] method and tables designated for determining the significance of the mean of a small series of paired differences and Fisher's modification[12] (p. 107) of Student's method for the comparison of two groups of unpaired data were used. Differences were considered significant when the probability (P) was less than 0.030. In addition, coefficients of correlation and correlation ratios were determined with the methods described by Ezekiel[13].

TABLE 2—ENVIRONMENTAL CONDITIONS AND OBSERVATIONS IN ROOM 1

ENVIRONMENT	ET	DBT F.	RH PERCENT	VP PSI	MT_s F	T_R F	THERMAL SENSATION	EWL[b] GM/MIN	PULSE RATE BEATS/MIN
I A	59.6	60.5	80	0.209	86.0	98.6	2.9	0.83	72.1
I B	59.6	63.4	30	0.087	87.1[a]	98.4	3.1	0.90	75.6
II A	68.8	70.3	80	0.293	91.5	98.3	3.6	0.73	74.9
II B	68.8	76.0	30	0.133	92.6[a]	98.4	4.0	1.08[a]	74.9
III A	73.4	76.0	80	0.355	92.7	98.3	4.1	0.82	76.9
III B	73.4	82.3	30	0.164	93.7[a]	98.3	4.6	1.17[a]	77.1
IV A	78.0	80.7	80	0.415	93.7	98.6	4.5	0.88	79.7
IV B	78.0	89.6	30	0.207	94.3	98.5	5.3[a]	1.78[a]	80.1
V A	82.6	85.9	80	0.491	93.7	98.6	5.9	1.85	83.7
V B	82.6	97.5	30	0.264	94.7[a]	98.6	5.9	3.00[a]	87.0

[a] Difference statistically significant for that effective temperature.
[b] Evaporative weight loss.

Results

Room 1. By the end of the hour of exposure to room 1, the subjects were in approximate equilibrium with the environment. The mean skin temperature (MT_S) had remained relatively unchanged during the final 30 min for all environments except that designated I A. Here it showed a slight decrease during the final 20 min. The averages of the data obtained for these 10 ambient environments are presented in Table 2.

The final mean skin temperature was significantly greater at the higher than at the lower dry bulb temperature for four of the five effective temperatures studied ($P = 0.004$, 0.020, 0.016 and 0.002 for environments I, II, III and V, respectively). For environment IV, the mean skin temperature was not significantly greater at the higher dry bulb temperature although it was in the same

Table 3—Range of Thermal Sensations and Number of Votes Recorded at the Different Ambient Environments in Room 1

Environment	2	2½	3	3½	4	4½	5	5½	6	6½	7
I A	2		3	2							
B		2	3	1	1						
II A			2	2	3						
B				1	5	1					
III A				1	4	1	1				
B					1	3	3				
IV A					2	3	2				
B						2	2		3		
V A							2	1	2	1	1
B							1	2	3		1

direction as already given. This indicates that two conditions having the same effective temperature will not necessarily have the same effect on mean skin temperature. This has been pointed out by Yaglou[8] for mean skin temperature and by Sheard[14] for forehead and extremity temperatures.

The average rectal temperature (T_R) was similar for the wide range of effective temperatures studied. If the subjects had remained in this room for a longer time at the coolest environment (zone of body cooling), a difference would have appeared.

The average pulse rate was similar for the two environments of the same effective temperature but the rate was greater at the high effective temperatures.

It had been observed[15] that evaporative weight loss was slightly greater during the first 10-15 min after the subject resumed his seat on the balance. This was attributed to the evaporation of moisture initially present in the union suit. Once equilibrium was established, the rate became relatively constant and was due to loss from the body. The data, therefore, were analyzed from the time the rate of weight loss became constant. It was found (Table 2) that for four of the five effective temperatures studied the rate of weight loss was definitely greater for the condition having the higher dry bulb temperature ($P = 0.012$, 0.025, 0.010 and 0.004 for conditions II, III, IV and V, respectively). For two

conditions having the same effective temperature in this environmental range, a greater evaporative weight loss was expected for the condition having the higher dry bulb temperature and lower relative humidity. This was also shown by Houghten et al[16].

Table 3 shows the distribution of votes recorded at the different ambient environments. The thermal sensations were not significantly different for two environments of the same effective temperature (Table 2) with one exception. However, for environments II, III, and IV the votes tended to be higher for the higher dry bulb temperatures. The high correlation ratio, discussed in the following paragraphs, between thermal sensation and dry bulb temperature and the absence of statistically significant differences between two environments of

TABLE 4—OBSERVATIONS IN HOT ROOM

ENVIRONMENT	Δ MT_s AT 10 MIN F	FINAL MT_s F	Δ T_R IN 10 MIN F	FINAL T_R F	THERMAL SENSATION		FINAL PULSE RATE
					INITIAL	FINAL	
I A	7.7	96.7	0.26	98.2	5.9	6.4	86.3
I B	7.9	96.8	0.26	98.4	6.3	6.4	84.0
II A	4.6	96.9	0.24	98.6	6.0	6.1	89.4
II B	4.2	97.0	0.08[a]	98.9	6.0	6.4	91.1
III A	3.6	97.0	0.28	98.6	6.5	6.4	91.4
III B	3.1	96.8	0.15[a]	98.7	6.3	6.4	91.4
IV A	2.7	97.3	0.11	99.0	6.2	6.6	93.1
IV B	2.4	97.0	0.03[b]	99.0	6.6	6.7	93.9
V A	2.3	97.0	0.13	98.9	6.6	6.9	93.4
V B	1.8	97.3	0.01[a]	99.1	6.8	6.9	101.1

[a] Difference statistically significant for that effective temperature.
[b] Difference statistically suggestive for that effective temperature.

the same effective temperature indicate that each group should contain a greater number of subjects before definite conclusions may be drawn.

Hot Room: In general, the major part of the redistribution of blood is accomplished within 10 min of entering the hot room. This was observed for environments II, III, IV and V. The rapidity and magnitude of the increase in mean skin temperature has been described[7]. The rapid increase was also noted for environments I A and I B but here the average mean skin temperature remained significantly lower than that of the other groups for at least the first 40 min. By the end of the hour, however, the average mean skin temperatures were approximately the same for all groups (Table 4).

The increase for the first 10 min in the hot room exhibited an apparent linear correlation with the level of mean skin temperature observed prior to entrance. The coefficient of correlation obtained was 0.82 $\pm$0.04 for a mean skin temperature range of 85 F to 95 F. This linearity may not be retained at lower and higher mean skin temperatures.

The decrease in rectal temperature within 10 min of entering the hot room was small but definite in eight of the 10 conditions (Table 4). Only in environ-

ments IV B and V B where the dry bulb temperature and mean skin temperature were the highest in room 1 did the rectal temperature remain unchanged. This effect on rectal temperature of sudden exposure to heat has been reported by this laboratory and others[7-9, 17-22] and is a result of the transfer of heat from the deep to the superficial tissues. For previous exposures to two environments having the same effective temperature, the decrease was slightly less for the environment having the higher dry bulb temperature and mean skin temperature, with the exception of environment I.

For all environments, except I A and I B, the final rectal temperature in the hot room showed an increase over the final room 1 value. The increases were not significantly different for each pair of environments having the same effective temperature. For I A the T_R decreased 0.4 F whereas no change occurred for I B. The difference was significant ($P = 0.010$). For these two environments in room 1 the mean skin temperature averaged at least 4.4 and 5.5 F lower than at any of the other conditions (Table 2). And, as mentioned, the mean skin temperature remained significantly lower than that of the other groups for the first 40 min in the hot room.

The average change in pulse rate from the end of room 1 to the end of the hot room (Tables 2 and 4) was similar for all environments. The final pulse rate in the hot room was greater when the subjects were previously exposed to the higher ambient environments of room 1.

The amount of perspiration which occurred in the hot room was estimated according to the formula previously presented[8]. In general, the subjects perspired more when the preceding exposure in room 1 was to the higher dry bulb temperatures.

The degree of warmth experienced upon entering the hot room (Table 4) was not significantly different for each of two environments having the same effective temperature and apparently bears little relationship to the thermal sensation, ambient environment, or mean skin temperature in room 1. If each group had contained a greater number of subjects, the initial sensation of warmth might have been significantly different for the two extreme conditions (I *vs.* V). The *change* in thermal sensation is, of course, related to the factors mentioned.

After one hour in the hot room, sensation of warmth was similar for all conditions, except that after a previous exposure to environment V the subjects felt definitely ($P = 0.016$) warmer than after exposure to environment I.

Room 2. (After hot room exposure): Upon subjects' re-entrance of this room, the mean skin temperature decreased rapidly (Fig. 1), as has been previously shown[7-9]. For the first 10 min, as well as for the hour, the change from the final hot room value was about the same for each environment having the same effective temperature, except for environment V. Here the decrease was slightly less at 10 min ($M = 0.7$ F; $P = 0.019$) and at the end of the hour ($M = 1.2$ F; $P = 0.004$) for the environment having the higher dry bulb temperature (V B). Also, for this environment only, the final mean skin temperature in room 2 was significantly higher for the higher dry bulb temperature. This suggests that under dynamic conditions, with perspiration present on the body and in the union suit, the effective temperature index is applicable for mean skin temperature changes with slight modification at the warmest environment (V). Here again it appears that less emphasis should be placed on relative humidity.

It is of interest to note the effect of RH upon the decrease in mean skin temperature during the first 10 min in room 2. A previous observation that the decrease in mean skin temperature was significantly greater at 30 than at 80 percent RH when the dry bulb temperature was 76 F[7] was confirmed. Further, by comparing the response obtained for environment III B (82.3 F and 30 percent RH) with that obtained for environment IV A (80.7 F and 80 percent RH) it was found that the decrease was slightly greater for III B. Similarly, in comparing environment IV B (89.6 F and 30 percent RH) with

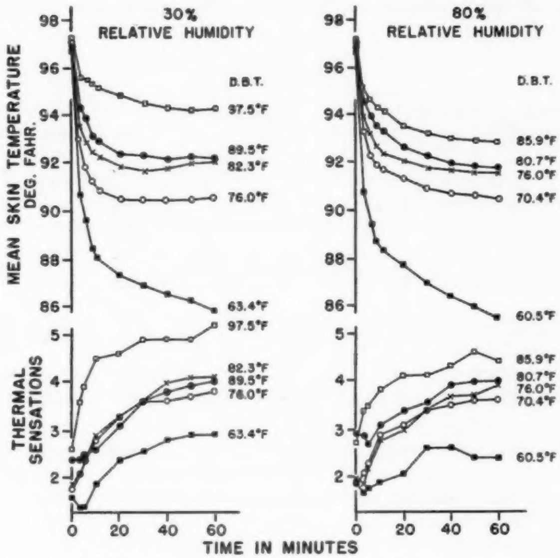

FIG. 1. MT_S AND THERMAL SENSATIONS IN ROOM 2 FOR THE 10 AMBIENT ENVIRONMENTS

environment V A (85.9 F and 80 percent RH) the decrease was definitely greater for IV B despite its higher dry bulb temperature.

Relative humidity also appeared to influence the change in mean skin temperature during the entire exposure in room 2. When all experiments at 30 percent were compared with those at 80 percent, the change in mean skin temperature was slightly less ($M = 0.6$ F; $P = 0.007$) for the 80 percent group. This held true even for the last 30 min in room 2 ($M = 0.3$ F; $P < 0.001$).

The rectal temperature showed a slight but definite increase within 10 min of entering room 2 for all environments. This has been discussed[7-9, 21, 22]. The magnitude of the change, however, was not affected by the effective temperature of room 2, since the increase was about the same for all environments. Nor was the change from the end of the hot room to the end of room 2 significantly different for each of two environments having the same effective temperature.

The first pulse rate was obtained about 3 min after entering room 2 and at regular intervals thereafter. The average change in pulse rate during the hour from the final value in the hot room was similar for the two environments of the same effective temperature. However, the final pulse rate was greater at the higher ambient conditions as was noted in the exposure to room 1 and in the hot room.

The weight upon entrance into room 2 could not be accurately determined. Therefore, the evaporative weight loss in room 2 was interpolated to a standard time (5 min) after the first weight. Table 5 presents the average evaporative weight loss for each ambient environment. The evaporative weight loss was

TABLE 5—OBSERVATIONS IN ROOM 2

Environment	Δ MT_s at 10 Min F	Final MT_s F	Δ T_R in 10 Min F	Final T_R F	Thermal Sensation					EWL[b] gms	Pulse Rate	
					Initial	3 Min	5 Min	10 Min	Final		3 Min	60 Min
I A	8.1	85.6	0.23	98.3	1.9	1.7	1.8	1.9	2.4	75.3	76.1	69.7
I B	8.5	85.9	0.23	98.3	1.6	1.4	1.4	1.9	2.9	104.1[a]	78.9	69.7
II A	5.2	90.4	0.19	98.3	1.9	2.1	2.3	2.9	3.6	94.4	77.1	72.9
II B	6.0	90.6	0.12	98.3	1.8	2.1	2.4	2.8	3.8	120.3[a]	77.4	72.6
III A	4.3	91.5	0.18	98.4	2.0	1.9	2.2	2.8	3.9	99.3	80.1	74.9
III B	4.4	92.0	0.13	98.3	1.9	2.1	2.4	2.9	4.1	135.0[a]	81.1	78.7
IV A	3.9	91.8	0.23	98.9	2.9	2.9	2.7	3.1	4.0	132.7	84.0	80.3
IV B	3.9	92.2	0.17	99.0	2.4	2.4	2.5	2.6	4.0	190.9[a]	84.3	84.9
V A	2.8	92.8	0.24	99.1	2.7	3.4	3.6	3.8	4.4	143.3	85.7	82.3
V B	2.1[a]	94.3[a]	0.20	99.3	2.6	3.6	3.9	4.5	5.2	204.4[a]	92.3	86.0

[a] Difference statistically significant for that effective temperature.
[b] Evaporative weight loss.

significantly greater at the lower than at the higher relative humidity for all of the effective temperatures studied even though the differences were small. Further, it should be noted that the amount of moisture available for evaporation upon entrance into room 2 was greater at the lower than at the higher relative humidity for each effective temperature studied.

Thermal Sensation: The averages of the votes recorded in room 2 for the 10 ambient environments studied are shown in the lower portion of Fig. 1.

The *initial* thermal sensation upon entering room 2, after the hot room exposure, was similar for the two environments of the same effective temperature (Table 5). This was to be expected since the *Effective Temperature Index* was predicated upon *instantaneous* thermal reactions. Further, the thermal sensations were not significantly different during the hour (3,5,10,20 min, etc.) for the two environments of the same effective temperature (Table 5), although some differences in means were present.

Since the thermal sensations for the two environments of the same effective temperature were not significantly different at comparable times during the hour these votes were grouped. The mean values obtained for each effective

temperature were then compared. This revealed that the *initial* sensation of coolness was approximately the same for environments I, II, III 59.6, 68.8 and 73.4 ET, respectively) even though the dry bulb temperature ranged from 60.5 to 82.3 F. The explanation for this lies in the fact that the experiments were conducted on different days and that the *subjects were not comparing one environment with another,* their votes expressed their thermal sensations at that moment. Within *three minutes,* however, the sensation of coolness was definitely less for environment II than for environment I ($P = 0.023$) and the subjects remained definitely less cool for the remainder of the hour. This was also true for the comparison of environment I with III, although the difference at 3 min was only slightly suggestive. The subjects were definitely cooler in environment I than in IV or V from the *moment of entrance* to the end of the hour.

Only when the subjects entered environment IV (78 ET) did the *initial* votes show a difference in the sensation of coolness experienced. Upon entrance into environment IV, the subjects definitely voted higher than for environment II ($P = 0.021$) and suggestively higher than for environment III ($P = 0.041$). For the remainder of the hour there was a striking similarity in votes for environments II, III and IV (68.8, 73.4 and 78.0 ET, respectively). It should be noted here that even though the average votes for environments IV A and IV B were higher than for environments II A, II B, III A or III B during the first 5 min in room 2 (Table 5) the differences between the means were not statistically significant.

When the subjects were exposed to the highest effective temperature studied (environment V), they were definitely warmer than in the other environments from the moment of entrance to the end of the hour, with one exception. Only for the *initial* vote in room 2 was there no significant difference between environment V and IV.

Correlation of Thermal Sensations with Mean Skin Temperature, Effective Temperature and Dry Bulb Temperature in Room 1: In a previous report[7], a discussion of the range of mean skin temperature at a time when subjects were comfortable was presented. In the present study, as a result of the wide range in ambient conditions, a greater range of thermal sensations has been obtained in room 1 in which the subjects were in approximate equilibrium with the environment. These data permit the calculation of the following correlation ratios for the entire range of ambient conditions studied.

	Correlation Ratio	Standard Error
Thermal Sensation and MT_s	+0.70	±0.06
Thermal Sensation and ET	+0.85	±0.03
Thermal Sensation and DBT	+0.86	±0.03

All three correlation ratios are highly significant. A prediction of thermal sensation, however, could be obtained with greater reliability with either dry bulb temperature or effective temperature than with mean skin temperature. Dry bulb temperature, of course, is the simplest to obtain. It should be mentioned, however, that at very high dry bulb temperatures and high humidities such a relationship would not hold.

Bedford[6] concluded that comfort could be predicted with slightly greater accuracy if the warmth of the room were known rather than the skin temperature of the individual. His conclusion, however, was based upon the use of only three skin areas, not mean skin temperature. Furthermore, two of the three skin temperatures were on the extremities.

Definition of Comfort and Discomfort. The term *comfort* is one part of the scale of thermal sensations as used by this laboratory and others. Actually, comfort is not a sensation in itself; rather, it is a *feeling-state.* Leopold[23] indicated

TABLE 6—SUMMARY OF OBSERVATIONS OBTAINED IN STEADY AND DYNAMIC STATES

OBSERVATION	ROOM 1 STEADY STATE	ROOM 2 DYNAMIC STATE
Mean Skin Temperature	−	+ (− for cond V only)
Rectal Temperature	+	+
Thermal Sensation	±	+
Evaporative Weight Loss	−	−
Pulse Rate	+	+

Note: + indicates agreement with effective temperature index.

the need for an adequate definition of comfort. The authors offer the following definition:

Comfort, in terms of thermal sensations to the physical environment, is a derived state of feeling based upon a physiological balance of the individual to his environment wherein the thermal stimuli are of low intensity.

Discomfort, in terms of thermal sensations to the physical environment, is a derived state of feeling based upon a physiological imbalance of the individual to his environment or a physiological balance wherein the thermal stimuli are of high intensity.

The intensity of the thermal stimuli required to effect a change in the feeling-state should be determined by research in the near future.

Correlation of Mean Skin Temperature with Effective Temperature and Dry Bulb Temperature in Room 1: A very high correlation ratio was obtained with mean skin temperature and effective temperature or dry bulb temperature.

	CORRELATION RATIO	STANDARD ERROR
MT_s and ET	+0.95	±0.01
MT_s and DBT	+0.96	±0.01

For predicting mean skin temperature over the range of ambient conditions studied, either factor is equally good and both are significantly better ($P < 0.001$) than thermal sensation (*comfort* vote).

Hardy and Du Bois[24] reported a high degree of correlation between mean skin temperature and dry bulb temperature for two nude male subjects under basal conditions. For every degree rise in environmental temperature the mean

skin temperature was about 0.5 F greater in the temperature range of 71.6 to 84.2 F.

Fig. 2 presents the distribution of mean skin temperature plotted against dry bulb temperature for all experiments in room 1. As mentioned, these data yield a correlation ratio of 0.96 ±0.01. The averages for thermal sensations expected at each dry bulb temperature have been placed at the upper portion of the graph. This was done merely to indicate the level of mean skin temperature at which different thermal sensations may be expected.

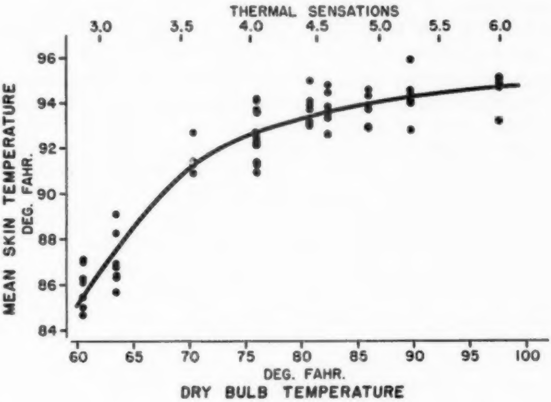

Fig. 2. Relation of MT_S to DBT with Thermal Sensations Expected

Table 6 summarizes the data presented in this paper. Observations in agreement with the effective temperature index are indicated by a plus sign and those in disagreement by a minus sign. For the steady state, *i.e.*, with subjects in approximate equilibrium with the environment (room 1), it is apparent that differences exist for mean skin temperature, evaporative weight loss, and probably for thermal sensation in the intermediate range. As measured by the effect on mean skin temperature, the effective temperature index places too much emphasis on the influence of relative humidity throughout the range of ambient conditions studied.

For the dynamic state, *i.e.*, in room 2 after an exposure to the hot room for one hour, the results were in agreement with the effective temperature index except for mean skin temperature at the highest effective temperature studied (condition V) and for evaporative weight loss at all effective temperatures. As measured by the response of mean skin temperature, relative humidity is overemphasized at this high condition. The similarity in results obtained in the dynamic state for two ambient conditions having the same effective temperature was not surprising. As mentioned earlier, the effective temperature index was developed with subjects in a dynamic state.

Discussion

Yaglou[3] suggested that lines of equal mean skin temperature be substituted for effective temperature lines on the psychrometric chart. This was based upon (*a*) the close relationship between mean skin temperature and thermal sensations in ambient conditions which are below the zone of evaporative regulation[1, 25, 26] and (*b*) upon data obtained in a few exploratory experiments on two subjects[3]. In this zone it is agreed that effective temperature overemphasizes the influence of relative humidity and that a high correlation exists between mean skin temperature and dry bulb temperature or effective temperature. However, the substitution of lines of constant mean skin temperature for effective temperature lines in that temperature range is of questionable value. Thermal sensations can be predicted with greater reliability by dry bulb temperature alone than by mean skin temperature alone, as noted. Therefore, the introduction of another variable which is more difficult to obtain requires greater justification.

This, of course, does not imply that for all environmental conditions the dry bulb temperature would be a better index of thermal sensation than mean skin temperature. It is highly probable that for non-uniform environments, *e.g.* where there is asymmetry of radiation, with subjects under more or less equilibrium conditions, mean skin temperature, as suggested by Yaglou[3], rather than dry bulb temperature would be a better measure of thermal sensations. Under dynamic conditions, it is probable that the rate of change of mean skin temperature correlated with the level of mean skin temperature would prove a better measure. This was mentioned earlier[7]* and such a study is in progress covering a wide range of ambient conditions. Addition of the depth of the tissue gradient to the rate of change of mean skin temperature and the level of mean skin temperature would probably further improve the relationship.

Conclusions

Fifteen healthy young adult male students, clothed in 90 percent cotton union suits and wooden sandals, were subjects for the experiments. Each of the subjects was exposed to two ambient environments having the same effective temperature, although only seven subjects comprised a group for study at each effective temperature. Effective temperature in the zones of body cooling, thermal neutrality and evaporative regulation were selected. Observations of skin and rectal temperatures, evaporative weight loss, pulse rate and thermal sensations were obtained.

The data justify the following conclusions for subjects in approximate equilibrium with the ambient environment (room 1):

1. The final mean skin temperature (MT_S) was significantly greater at the higher than at the lower dry bulb temperature (DBT) for two ambient environments of the same effective temperature.
2. The rectal temperature (T_R) was similar for the wide range of effective temperatures studied.
3. The pulse rate was similar for the two environments of the same effective temperature but the rate was greater at the higher effective temperatures.
4. The rate of evaporative weight loss was definitely greater at the higher than at

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* Authors' Closure, A.S.H.V.E. TRANSACTIONS, Vol. 53, 1947, p. 356.

the lower dry bulb temperature for two ambient environments of the same effective temperature.

5. Thermal sensations were not significantly different, although they tended to be higher at the higher dry bulb temperature, for two ambient environments of the same effective temperature.

6. Thermal sensations were correlated with effective temperature, dry bulb temperature, and mean skin temperature. All three correlation ratios are highly significant.

7. A definition of comfort and one of discomfort are offered.

8. Mean skin temperature correlated very highly with effective temperature and dry bulb temperature.

The data justify the following conclusions for subjects in a dynamic state, *i.e.,* in room 2 after one hour in the hot room:

1. The decrease in mean skin temperature was similar for two ambient environments of the same effective temperature, except for the highest effective temperature studied.

2. The changes in rectal temperature and pulse rate were similar for two ambient environments of the same effective temperature.

3. The evaporative weight loss was significantly greater at the lower than at the higher relative humidity for all effective temperatures studied.

4. Thermal sensations were similar for two ambient environments of the same effective temperature.

In general, the results suggest that the present effective temperature index is adequate for subjects in the dynamic state (room 2) except for correction at the higher temperatures. For subjects in a steady state (room 1), the present effective temperature index places too much emphasis on the influence of relative humidity for the range of ambient conditions studied.

ACKNOWLEDGMENT

The authors gratefully acknowledge the valuable assistance of Dr. Stanley E. Telser and Miss Jean Shaw.

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9. Physiologic Adjustments of Normal Subjects and Cardiac Patients to Sudden Change in Environment, by Nathaniel Glickman, Tohru Inouye, R. W. Keeton, M.D., I. R. Callen, M.D., F. K. Hick, M.D., and M. K. Fahnestock. (A.S.H.V.E. TRANSACTIONS, Vol. 55, 1949, p. 27.)

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DISCUSSION

LESTER T. AVERY, Cleveland, Ohio (WRITTEN): The study of man and his relationship to his air environment has been going on for centuries. Our Society has identified itself with this work for years, and will continue to do so.

With reference to this paper I appreciate the restraint with which the authors state *that for equilibrium conditions the present Effective Temperature Index places too*

much emphasis on the influence of relative humidity. After studying the paper and reading between the lines in the conclusion, it would seem to me that we find that the effective temperature lines are possibly a little too refined and that, therefore, we need effective temperature zones.

Here we cross into the field that C. S. Leopold and others have investigated and found that in the general range of 20 to 60 percent RH and the general range of 69 to 71 ET it is hard to choose between any one point and another. Possibly we need a group of *effective temperature* zones, starting with the one that is considered most favorably or the one in which there is minimum strain. Then the two border zones below and above would be determined and described as slightly too cool and slightly too warm. Then would follow that broad pair of zones, cold or hot. Then the zone of freezing and below on one side, and fever heat on the other, could be used to complete the picture. Maybe we have been trying to over-refine a general condition and in so doing have lost sight of the important thing, which is that there is a relationship between the effective temperature that is comfortable to most people and the zone of minimum strain which the physiologist mentions.

It is my hope that in the re-study of effective temperature, at our Research Laboratory under the guidance of the TAC Sensations of Comfort we will get a better tool to work with so that the engineer, the physiologist and the medical practitioner can meet on a common ground and agree on the design conditions and on the general range of temperatures and humidities to be permitted within this condition. Let's take the next step forward.

C. M. Ashley, New York, N. Y. (Written): This paper contains some very valuable data and deserves very serious and careful study.

While the effect of going from relatively comfortable to hot environments and back again is itself of considerable interest, I found the principal point of interest to be the data shown in Table 2 for room 1 under the relatively comfortable conditions. The data indicates clearly that there are two distinct temperature zones: the lower in which the dry bulb temperature is the predominant factor, and the upper, above the sweat line, where the effective temperature is the predominant factor.

The data of Table 2 for mean skin temperature, thermal sensation index and body moisture loss have been plotted *vs.* the dry bulb temperature. The values of mean skin temperature at 80 and 30 percent RH correlate remarkably well on a dry bulb temperature basis; however, the much higher mean skin temperature shown in the hot room suggests that there is a deviation between the high and low relative humidities which, unfortunately, is not shown within the range of the observed readings in room 1.

The thermal sensation index also correlates remarkably well on a dry bulb temperature basis up to the highest temperature reading at 80 percent RH and the highest three temperature readings at 30 percent RH. On the other hand, the correlation between effective temperature and the thermal sensation index is very poor except at the top effective temperature.

Lest it be thought that the failure of correlation of the top temperature for which 80 percent relative humidity is shown is an accident, I have also plotted curves of dry bulb temperature *vs.* latent heat per person. First, it will be seen that the value of the latent heat for the lower temperature range is nearly constant and that there is only a relatively small difference between the rate of 80 and 30 percent RH, thus lending additional weight to the subjective observation that relative humidity has very little effect and that temperature is the predominant factor in controlling thermal sensation.

Again the break in the line corresponding to the sweat point occurs at approximately the same dry bulb temperature as the break between the 80 and the 30 percent RH lines for thermal sensation. It is interesting to note that the latent heat lines correspond reasonably well with those on page 218 of The Guide 1949 from the *Pierce Laboratory* data but do not correspond to the A.S.H.V.E. data.

From these data you could draw the conclusion that in the lower temperature ranges explored in this study the effective temperature lines should be almost vertical and should show an increasing slope as the temperature increases above the sweat line. These observations are, after all, entirely reasonable since it would be expected that where the skin was quite dry and little evaporation occurred, humidity would have correspondingly small effect; whereas humidity would become important when the temperature range was reached in which bodily temperature control was effected through evaporation from the body surface.

While the test appears to have been made with great care and attention to the niceties of experimental technique, the validity of the data is to some extent impaired by the fact that different experimental subjects were used for the different sets of readings. I think it would be of considerable value to study the individual records and attempt to adjust these wherever indicated for individual differences between the various subjects.

Another point of some importance is the fact the observations in room 2 did not return completely to the similar conditions for room 1. It would be interesting to know to what degree this lack of return is due to lack of complete stabilization and to what degree to a relatively long term effect on the subjects due to the hot room environment.

The present study would point to the need for further detailed exploration in the region and above the sweat point, and also for a substantial modification of the comfort chart.

DR. THOMAS BEDFORD*, London, England (WRITTEN): I have read this paper with much interest. The findings which are of especial interest to me are those concerning the inter-relationships between skin temperature, environmental warmth, and thermal sensations. As the authors have remarked, fourteen years ago I showed, on the basis of over 2,500 observations, that subjective feelings of warmth were predicted with greater accuracy from a knowledge of the warmth of the environment than from measurements of the skin temperature on three sites, namely, the forehead, a hand and a foot, and it is therefore of interest to me to read that a similar result is obtained when the mean skin temperature is measured.

I find it of some interest, also, that in the room 1 data (Table 2) there is a remarkably linear relationship between thermal sensations and dry bulb temperature, ignoring humidity, even when the temperature is as high as 97.5 F. The one divergent value is that for environment V A. The correlation ratios for thermal sensation on effective temperature and on dry bulb temperature are virtually identical, although the ratio for dry bulb temperature is slightly the greater. Yet in spite of this the data in Table 2 suggest to me that in these experiments dry bulb temperature was distinctly better than effective temperature as an index of warmth—at any rate as an index of the average sensation level for a group of people.

In these studies of reactions to the thermal environment it is undoubtedly desirable that measurements of skin temperature should be made when possible, for they throw light on physiological regulation, but when it comes to predicting what thermal sensations will be in given environments I have sympathy with the view of the authors, that further justification is needed before mean skin temperature is used as the index. Bluntly, if you want to know whether a man is comfortable or too warm, ask him, and do not tell him he should be comfortable because his skin temperature (according to your equation) says so.

A. R. BEHNKE, Bethesda, Md. (WRITTEN): In physiological studies aboard ship, we have found that the Effective Temperature Index has been extremely valuable, providing that some correction is made for the factor of radiant heat—a point which has been made by others in this connection. In the Navy, we have been very much inter-

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TABLE A—AVERAGE VALUES 10 MEN OBSERVED DAILY IN A COMPARTMENT OF A WARSHIP

DATE HOUR	SCHNEIDER INDEX	PULSE RATE SITTING	TEMPERATURE		AIR TEMPERATURE			REMARKS
			Oral	C	D.B.	W.B.	E.T.	
	(1)	(2)	(3)	(4)	(5)	(6)	(7)	
6/3								
1	10.7	70	98.9	88.6	78.5	69	74.5	Comfortably cool LIMA,
2	10.6	70	98.8	89.9				PERU
3	11.4	67	98.8	89.7				
6/4								
1	8.8	75	99.1	91.9	83	73	78	Comf or comf warm,
2	8.6	74	99.1	92.5				awake. Sea T-68
3	9.5	71	98.9	92.2				
4	9.9	68	98.9	92.1				
6/5								
1	7.6	77	99.4	91.7	83	73	78	5 men—comf and awake
2	8.6	72	99.1	91.6				3 men—comf and alert
3	9.5	69	98.9	92.1				2 men—comf warm
4	9.9	66	98.8	91.3				Sea T-70
6/6								
1	8.1	74	99.2		87	80	83	Out of Humboldt current
2	8.2	72	99.0					6 men comf warm, fore-
3	9.2	69	98.9					heads damp
4	9.5	68	98.8	92.6				4 men too warm, sweat- ing, Sea T-83
6/7								
1	5.7	76	99.1		91	81	85	
2	6.1	74	99.1					3 men hot, sweating
3	6.9	71	99.0					4 men too warm
4	8.1	68	99.1	93.6				2 comf warm
6/8								
1	7.2	76	99.2		89	80.5	84	8 men comf warm
2	7.6	73	99.1					2 men hot
3	8.8	69	99.0					
4	8.5	69	99.0	93.6				
6/9								
1	5.7	77	99.3	93.7	91	81	85	4 men hot, sweating
2	6.4	75	99.1					3 men too warm
3	6.4	73	99.0					2 men comf warm
4	7.2	70	99.0					Sea T-83
6/10								
1	8.7	73	98.9	93.1	87	76	81	AIR CONDITIONED
2	9.3	71	98.9	91.8	84.5	70	77	Sea T-68-70
3	9.6	67	98.8	92.0	84.5	68	76.5	Men comf or comf cool
4	11.0	65	98.8	91.6	83.0	65.5	75	
6/11								
1	10.4	69	98.9	90.2	80	68	74.5	Sea T-62-63
2	11.2	67	98.6	90.6				Men comf cool
3	12.4	63	98.5	90.4				
4	11.9	60	98.4	89.1				LONG BEACH, CALIF.

ested in the effect of relative humidity values above 80 percent, as well as the effects of extremely dry environments, relative humidity below 30 percent. Under these conditions the effective temperature line will perhaps not be straight but it will be *S* shaped at each end. From data obtained aboard ships operating in tropical waters with men either acclimatized or undergoing acclimatization, dry bulb temperatures in the

range of 77 to 83 F with respective relative humidity values of 80 to 50 percent are associated with heat lost from the body without visible sweating; that is to say, about 60 percent of the heat lost would be brought about by radiation and convection and 20 to 40 percent by the evaporation of insensible perspiration. In a submarine compartment, for example, a dry bulb temperature of 85 F associated with a relative humidity of 76 percent defined an air environment that produced sweating of personnel. On the other hand, a somewhat lower dry bulb temperature of 83 F and a relative humidity of 73 percent, did not include sweating. In a compartment of a battleship, men acclimatized to tropical temperatures were comfortable when the dry bulb temperature was 85 F and the relative humidity was 56 percent. However, it was noted that a dry bulb temperature of 85 F, and humidity values in the range of 65 percent or higher resulted in damp and sweating skins among the personnel in the compartment.

With respect to the concept of Effective Temperature, values of 77 and 78 ET (30 to 80 percent RH) defined temperature and humidity conditions which are the upper limit with respect to the initiation of sweating. The following table presents data which illustrate the use we have made of the *effective temperature* concept in our test work during the past years. Under Column C the temperatures are those of the air layer next to the skin over the sternum. This envelope temperature, I believe, is very important and it would be of interest in future tests to make some attempt to measure the temperature of the air layer at a distance of one-quarter of an inch from the skin, using shielded thermometers or some other method for temperature measurement.

C. S. LEOPOLD, Philadelphia, Pa. (WRITTEN): According to the original effective temperature data for summer conditions, a change from 35 to 60 percent RH represents approximately a 3 deg change in dry bulb temperature. Practically all investigators agree with the general effect of relative humidity, but disagree slightly as to whether the effect is 1½ deg or 3 deg in the range of the 35 to 60 percent at 71 ET in summer.

Prof. Glickman states that the original data were obtained from instantaneous thermal impressions, but I think the record will show that as early as the end of 1923 the laboratory substantially verified these data for longer exposures.

One quoted paper by Rowley et al showed a reverse in effect of relative humidity, but I believe that this analysis was based on test data obtained with improper procedure by other investigators. My discussion of this report may be found in the A.S.H.V.E. TRANSACTIONS, Vol. 53, 1947. Prof. Glickman, however, in this and other preceding papers has ably confirmed the general effect of relative humidity.

In appraising the results of this paper for its application in everyday practice, consideration should be given to several points: *First,* the union suit used in this experiment is not demonstrated to be the same as multi-layer clothing as to radiation, convection and moisture retention effects. *Second,* of equal or greater importance, all subjects in this experiment wore the same type of union suit, whereas in ordinary living there is a wide diversity in clothing.

In a previous paper I tried to explain why it was necessary to keep such close tolerance in comfort air conditioning in spite of the fact that it takes approximately 7½ deg to change one subject from cool to warm. This effect is diagrammed in Fig. A in which large temperature tolerance is credited to each subject, but the range is shifted in an anticipation of the effect of clothing and physiological variables of the individual.

It could very well be that Prof. Glickman will define a new line for individuals identically clothed that would have little relation to an ordinary group dissimilarly clothed.

I am glad that Prof. Glickman agrees that we are in need of a new definition for comfort. In a paper on this subject I chose: "The absence of discomfort or annoyances due to temperature and atmospheric effects indoors." I believe that this definition can be improved, but I prefer the general approach to that proposed by Prof.

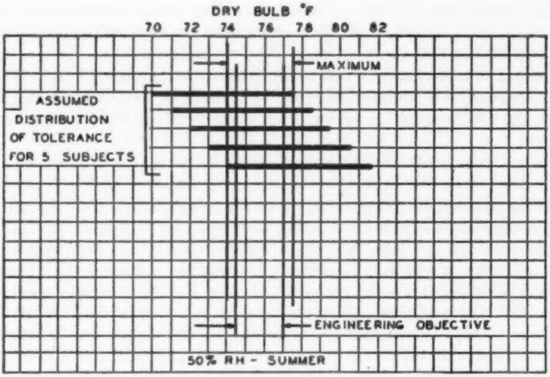

FIG. A. ILLUSTRATION OF RELATION OF GROUP TO INDIVIDUAL TOLERANCE

Glickman in which he refers to stimulus. Stimulus is difficult of definition and can lead to the confusion that has existed in much of the previous literature in which investigators try to find what a man can stand without excessive discomfort, rather than what a group would like if they did not have to pay for the results. Regardless of the physiological effects, if a man thinks he's cold, he is cold; if he thinks he's hot, he's hot.

DR. W. J. MCCONNELL, New York, N. Y. (WRITTEN): The authors of this paper have contributed pertinent information in the further evaluation of the *effective temperature index*. Attention may be called to the fact that the effective temperature index was not wholly developed from instantaneous thermal impressions of subjects passing back and forth between conditioned rooms. There were a series of three groups of tests where: (1) twelve experienced subjects remained in the test chamber for a period of three hours, from 9:00 to 12:00 in the morning, and two and a half hours, from 1:00 to 3:30 in the afternoon; (2) fourteen subjects participated in a series of two-hour tests; and (3) 100 individuals participated in a series of short tests of fifteen minutes or longer.

In addition to these short series, physiologic studies of the effects of high temperatures and varying humidities were conducted in a series of studies over a period of years.

The difficulty encountered in determining a single index that would indicate the degree of warmth perceived by an individual under various environments lies in the complexity of physical and physiological factors affecting thermal comfort.

C. L. TAYLOR*, Los Angeles, Calif. (WRITTEN): The authors are to be commended on a technically sound and competent investigation of the validity of the effective temperature scale with regard to temperature-humidity equivalences. Various imperfections in the effective temperature scale have from time to time come to light, but most common has been the finding that, at least in the approximate range 60-80 ET, the negative slope of effective temperature on a temperature-humidity plot is too great. The present paper confirms this fact, and further emphasizes the incompatibility on evidence from steady state and transient physiological responses.

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* Engineering Research, Department of Engineering, University of California.

Incidentally, would not the term, transient, rather than dynamic, be more suitable in this context? It is standard in the heat transfer terminology, while dynamic is most commonly understood in relation to Newtonian mechanics.

The redefinition of thermal comfort and discomfort is timely and the incorporation of the concept of physiological balance or imbalance as the basic factor determining the intensity of thermal sensation and feeling is worth the emphasis given it by the authors.

Two additional comments on the paper occur to me: (1) a more positive contribution would have been made if the results were adduced to show what correction factors might be applied to the present effective temperature scale to afford a modified effective temperature more compatible with their results and (2) the practice of testing at two ends of the humidity scale wholly disregards the possibility that the shape of the effective temperature curves may explain many of the apparent contradictions. Hall, Marbarger and I obtained data in hot room experiments at Wright Field during 1945 (unpublished) showing that lines of equal physiological effect are not linear. This is also shown in the work of Dr. Sid Robinson (*American Journal of Physiology*, 143:21 (1945)) and in the investigations of Dr. D. H. K. Lee (personal communication). Generally, these lines have an upward inflection, that is, at low and moderate humidities, tolerance contours follow dry bulb temperature lines, but approaching saturation they bend upward as the humidity effect asserts itself. This suggests that the humidity effect upon thermal comfort and balance is minimal until a critically low evaporative potential is reached, then impedance to heat loss causes the thermo-mimetic effect. In further investigations, exploration along the humidity line may considerably add to our knowledge of the true temperature-humidity relationship with either thermal sensation or physiological effect as parameter.

C.-E. A. WINSLOW, New Haven, Conn. (WRITTEN): The Society is to be congratulated on this admirable paper and the many other contributions coming from Dr. Keeton's Laboratory.

We have all of us, I think, been somewhat concerned as to the accuracy of the *effective temperature* in the middle range of human comfort. Dr. Herrington and I have stated on this point:‡ "It has been realized for some time that the scale provides a very satisfactory index of equivalent conditions at dry-bulb temperatures above approximately 81 F. In the lower comfort range, the scale likewise serves to equate the equivalent sensation effects of humidity and temperature for contrast situations in which a person passes from a moderate dry-bulb atmosphere to a similar temperature at high (or low) humidity. After adaptation over a period of one to two hours, however, the sensations realized are not in complete accord with the scale. In this moderate temperature range (68-81 F) the effect of humidity is overemphasized."

We have also long urged that in this area mean skin temperature is the true physiological index of comfort.

These suggestions, and the earlier observations of Yaglou[Δ] have been verified by the careful study presented today. The casual reader of this paper might be deceived by the fact that the effective temperatures and dry-bulb temperatures are both equally —and very closely—correlated with mean skin temperature. But correlation here measures merely a trend. The actual figures in Table 2 for the subjects in equilibrium with the environment show that with the same effective temperature, an increase of only 3 deg in dry-bulb temperature may produce a rise in mean skin temperature of 1 deg. This represents a highly significant effect on comfort.

The time would seem ripe for conducting sufficient studies of this kind to make possible a fundamental revision of the Effective Temperature Chart. The steady state

‡ *Temperature and Human Life*, Princeton University Press, 1949.

Δ A Method of Improving the Effective Temperature Index, by C. P. Yaglou (A.S.H.V.E. TRANSACTIONS, Vol. 53, 1947, p. 307).

is clearly the one for which air conditioning must be designed, not the contrast effect experienced in passing from one environment to another.

W. L. Fleisher, New York, N. Y.: Probably everybody knows that this is a subject which I have been particularly interested in for a good many years. Whenever we approach this subject we seem to establish some narrow limitations in our discussion of the subject and in our research investigations.

I take particular exception to the statement in the conclusions which says that *for subjects in a steady state room, the present Effective Temperature Index places too much emphasis on the influence of relative humidity.* It is dogmatic expressions like these that have created such misapprehension in the minds of the general public. In this particular paper the relative humidity referred to is within limits seldom approached in the summer and has nothing whatsoever to do with winter conditioning. Nevertheless, people being very sensitive to the words *relative humidity* will feel, after reading the conclusions, that humidification in wintertime is not essential and that the moisture content of the air is of little importance at any time. This is caused by misuse of the words in these papers in discussing the effect of relative humidity under conditions which never exist in the wintertime. In the paper there is no differentiation made between actual practice, winter practices and laboratory practices.

I was very closely associated with the research which developed these particular points re-investigated by Professor Glickman; and there is one point not brought out on which I have suggested many times in these meetings that re-investigation should be made.

In the course of investigations made with practical groups, not laboratory groups, over a considerable number of years in many locations and in many buildings, it was indicated that there was an effective temperature beyond which nobody was comfortable. This point was on a 78 effective temperature line, the 78 effective temperature line having been determined in the usual laboratory practical method.

In our laboratory investigations of high conditions, I felt, as I have emphasized in many former meetings and in many writings, that the results were improperly arrived at—the curves on which conclusions were based have been determined by extrapolation. Actually, it should be obvious that one cannot predict to what temperature a human body would rise simply by trying to place it on a curve which runs through two other points having absolutely no connection with the particular human being.

First, as you remember, about 1923 Dr. McConnell investigated the effect of high environmental conditions with the idea of finding out at what particular effective temperature physiological changes took place which might be injurious. For some reason or other, we decided that a rise in temperature of 1.5 F deg was a dangerous condition and that we must, consequently, keep our conditions below the point where such a rise in body temperature took place. We determined on 87 ET but we only arrived at this temperature by extrapolation. People were removed from the test chambers when they showed a rise in body temperature of 1½ deg above the so-called normal of 98.6 F even though some people entered the chambers with a temperature 1½ deg above normal. This was a ridiculous procedure as these people may have established equilibrium at the same elevated temperature as the other subjects. I objected to this procedure and feel that it affected practically all of the curves that were developed on this basis.

As I said, the only thing that we really had which was definite was that nobody was comfortable above a 78 ET. These investigations were made in five different cities and also in the Metropolitan Life Insurance Company's building where Dr. McConnell was in charge of the experiments. However, my feeling is that we have never arrived at a proper method of investigating sensation.

I remember very definitely that Dean Seeley and I were at a meeting of physicians

in New York where Dr. DuBois gave a speech very similar to the one presented today in which he said that they were not able to measure any difference in human beings subjected to different relative humidities at temperatures below 82 F and that consequently, relative humidity was of no importance in this temperature range. Nevertheless, when I suggested to him that the thermometer was not nearly as accurate as a feeling of discomfort he withdrew his conclusions. What he has substituted since I have no means of telling.

There probably is an efficient or a possible measure of determining electrically what is uncomfortable and how the sensations of people change with a change in relative humidity. As we grow older and bigger, and I refer to the research of the Society, our methods should become more modern and less crude. The thing that I am emphasizing at this particular time, however, is that it is a mistake for us to go before the public with even the slightest emphasis on the negligibility of low relative humidities in the winter time when we haven't the slightest research data to indicate that it is not important. We are aware that winter conditions of over 30 percent RH are taboo because of the ineffectiveness of present vapor barriers against this condition or higher relative humidities. Notwithstanding the pressure of the insulating groups against the desirability of rather high relative humidities, for the sake of health and comfort we should not allow a paper to emanate from this Society which creates misapprehension on the part of the public.

AUTHORS' CLOSURE: First of all, we wish to thank all of the individuals who have presented written as well as oral discussions of the paper. I will try to cover a few of the points mentioned.

We are aware of the experiments to which Dr. McConnell refers. The particular paper which reported tests under the three conditions was written by C. P. Yaglou* in 1923. Results were not used in the derivation of the effective temperature index, but were used partly to verify the established effective temperature index and partly to determine the comfort zone on the effective temperature scale.

Dr. McConnell rightfully stresses the difficulty in determining a single index that would indicate the thermal sensation of an individual under various environmental conditions. Dry bulb temperature and effective temperature are useful to a certain extent, but neither include the factor of radiation.

We must remember that most of the research has been conducted in uniform environments where air and all wall temperatures were about equal. Actually, we live in a non-uniform environment, and in such an environment dry bulb temperature would probably not be as useful an indicator of comfort as it was in our experiment. A much greater amount of research is needed in a controlled non-uniform environment. We can certainly develop a method for reflecting thermal sensations in any event and also expect to be able to define that environment.

Mr. Leopold comments on the early work of the Laboratory in establishing the effective temperature index, and my reply to Dr. McConnell's comments on that point are in agreement. I agree with him as to the differences due to the union suit and the clothing that individuals wear. We anticipated this and have already completed a series of experiments in which we have used subjects dressed in normal or average summer weight clothing, such as tropical worsted suits. The test conditions were the same as in previous studies when the subjects wore union suits. Results will be reported in a paper being prepared. We probably will not find too great a difference in the two series of tests.

Mr. Leopold obtains his definition of comfort by a negative approach. He says it is the absence of discomfort. We prefer to use the positive approach. Discomfort and comfort are really just descriptive terms. They are not sensations.

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* Determination of the Comfort Zone, by F. C. Houghten and C. P. Yaglou (A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923, p. 361).

He stated that stimulus is difficult to define. If you use a dictionary, you find a very good definition which applies to our particular situation. That is: *any agent or environmental change capable of exciting a specific end organ of sensation,* and here in the skin we have the end organs for heat and cold. You find the definition in Webster's.

Dr. Lee has commented at an earlier date that readers of this paper in the field of physiology are not accustomed to vapor pressure expressed in terms of pounds per square inch. This paper, however, was prepared for an engineering journal; and since we have used temperatures in degrees Fahrenheit, and relative humidity, we should be consistent and use pounds per square inch. Dr. Lee's suggestion of inserting the conversion factor for expressing vapor pressure in mm of Hg was a very good one, however, and we accept it; a footnote has been added in the paper.

Dr. Lee stated to the authors that in dealing with man the term vapor pressure is preferred to relative humidity. I agree; however, we were not set up to present it that way; and whether we analyze our results in terms of vapor pressure or relative humidity, our conclusions would have been the same.

Mr. Ashley's points are well taken. With regard to the higher mean skin temperatures observed in the hot room, we do not feel that they should be compared with the mean skin temperatures obtained in Room 1 at the higher dry bulb temperature. We have stated in an earlier paper, giving the details of the rooms, that the hot room was maintained within ±1 F deg in temperature and within ±4 percent in relative humidity, whereas Room 1 was maintained within ±½ F deg and ±2 percent. Anyone wanting to make such comparisons should repeat these experiments with subjects in Room 1 at the same dry bulb temperature and relative humidity which we had in the hot room.

Mr. Ashley mentioned that the validity of the data was to some extent impaired because different experimental subjects were used for the different tests. Examination of Table 1 in the paper will show that although each subject did not participate in the experiments at *all* the effective temperatures, each subject did participate in *two* different conditions having the same effective temperature. The table, therefore, presents results which can be compared. Each subject can be compared with himself under two different conditions having the same effective temperature.

Mr. Ashley has called attention to the fact that some of the final results in Room 2 were not the same as found in Room 1. This is easily explained. The primary reason was that the union suit contained a greater amount of moisture at the end of exposure in Room 2 than it did in Room 1, and only one hour in Room 2 was insufficient for stabilization.

Dr. Taylor raised the question that the term *transient* rather than *dynamic* would be more suitable in the context. *Transitory* might be even better.

Dr. Taylor raised two points. In reference to his second point, a complete analysis of the problem would necessitate additional experiments in the middle ranges of relative humidity, particularly at the high temperatures. However, most of the effective temperatures used in this study were not high. The effects suggested by Dr. Taylor would probably be noted.

With regard to the first point, I do not feel that we at this time have sufficient data to determine quantitatively the correction factor for the effective temperature scale. Further, it is probable that research in controlled conditions of non-uniform environments may soon be possible in the new test rooms at the A.S.H.V.E. Research Laboratory. Then, entirely new effective temperature scales will probably be developed which will include the factor of radiation.

Dr. Bedford's comments have given me the opportunity of again pointing out that in a uniform environment the dry bulb temperature is best for predicting thermal sensation; while in non-uniform environments the case is not so simple. Under dynamic conditions or transitory conditions, there are a number of physiological factors that must be considered in the determination of thermal sensation. We must

keep in mind the level of the mean skin temperature, the rate of change of the mean skin temperature, and the temperature gradient of the tissues.

I agree with Mr. Fleisher that the statement in the conclusions might be misinterpreted if the purpose of this study is not understood. Further, additional work should be done at the lower relative humidities which obtain in winter.

We still do not have a good method for measuring thermal sensations but we are approaching one.

Mr. Avery's comments were very appropriate. We can certainly join in his hope that future studies at the A.S.H.V.E. Research Laboratory will yield the answers to many provocative questions.

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No. 1381

THERMODYNAMIC CRITERIA FOR HEAT PUMP PERFORMANCE

By John F. Sandfort*, Ames, Ia.

The development of the heat pump as an apparatus for heating buildings comes as a resurgence of an idea first introduced by Lord Kelvin in 1852. His proposed design[1] was a two cylinder *thermodynamic engine*, operating on an open air cycle, capable of both heating and cooling small buildings. For various reasons this engine was never built and the heat pump idea lay dormant for nearly 80 years.

During this period the refrigeration industry grew steadily and the early air cycles were gradually abandoned in favor of the more practical vapor compression cycles. When interest again began to develop in the heat pump, it was a natural step to adapt existing refrigeration equipment for this new application. This probably explains why, with rare exceptions, commercial heat pump development to date has concentrated on the single stage vapor compression cycle as the basis of design. However, some air cycle heat pump equipment has been developed in Switzerland and others have been proposed in this country[2]. Some United States aircraft are now air conditioned with air cycle equipment.

Of utmost importance in the design of any heat engine, heat pump, or refrigeration equipment is the establishment of standards against which the performance of any actual or ideal engine or cycle can be compared.

In heat engine design, the desired effect is a conversion of heat to work. This conversion, expressed as a percent, is the overall thermal efficiency of the heat engine. Thermodynamics teaches that the maximum conceivable thermal efficiency of any heat engine operation between two fixed temperature levels is equal to that of the classical Carnot engine.

In the case of heat pumps and refrigeration equipment, however, the concern is with a flow of energy from a low to a high temperature level rather than a conversion of energy from one form to another. Common experience, as expressed in the Second Law of Thermodynamics, reveals that this process cannot occur spontaneously, and therefore additional energy (usually, but not necessarily, work) is required to make the heat pump function.

With refrigeration equipment the desired effect is the transfer of heat from a low temperature area, while with a heat pump the desired effect is the transfer of heat to a high temperature area. For such equipment thermal efficiency has

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[1] Exponent numerals refer to Bibliography.

Presented at the 56th Annual Meeting of The American Society of Heating and Ventilating Engineers, Dallas, Tex., January 1950.

no meaning, and so the expression coefficient of performance has become the accepted method of indicating excellence of performance. Coefficient of performance, abbreviated (CP), can be defined as the ratio of the effect desired over the work required to produce that effect.

In the case of the heat pump this ratio must obviously always be greater than unity if any particular system or cycle is to have any value as a heating machine. But the question is: How large theoretically can this coefficient of performance be for any assigned condition of operation? This question is of particular importance in heat pump design since the heat pump is competitive with the various combustion heating systems.

From thermodynamics it is seen that if the Carnot engine is reversed the result is a heat pump in which the work input is an irreducible minimum for the amount of heat being transferred, and therefore the Carnot coefficient of performance has come to be regarded as the ultimate standard for heat pump design.

Inasmuch as the Carnot is an ideal unattainable cycle, the vapor compression cycle has been developed as the nearest practical approach to it. Studies have

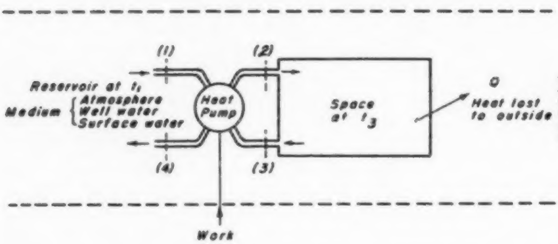


FIG. 1. SCHEMATIC DIAGRAM OF THE STEADY FLOW HEAT PUMP

appeared in the literature comparing the Carnot coefficient of performance with ideal vapor compression cycles and with actual commercial compressor designs[3, 4]. There has been speculation as to the probable future improvements along these lines. What has been generally overlooked, however, in evaluating maximum theoretical heat pump performance is the method by which it has been found necessary to employ these cycles in the use of space heating equipment.

It is the purpose of this paper to analyze such an apparatus thermodynamically, and through the concept of availability, determine the maximum conceivable coefficient of performance regardless of any particular cycle. This will provide a simple method for evaluating a standard against which the various actual or ideal cycles can be compared.

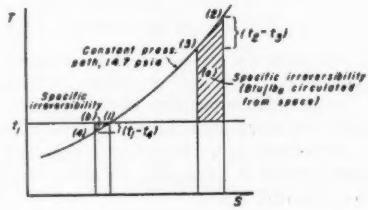
THE STEADY FLOW HEAT PUMP

Thermodynamically, a heat pump is an apparatus for causing heat to flow from a low temperature energy source or reservoir to a higher temperature space. The natural reservoirs available on earth are the atmosphere, surface water such as rivers or lakes, well water, and the earth itself. In certain situations artificial or man-made reservoirs such as low temperature waste heat from industrial processes can be used.

Experience has shown that the most practical way to operate a heat pump for space heating is by some steady flow process, schematically shown in Fig. 1. The usual method of extracting energy from the reservoir is to pass the medium of the reservoir itself through the heat pump in steady flow, in which case it would be rejected back to the reservoir at some lower temperature. An obvious exception would be the use of the earth as a reservoir. In this case a refrigerant or other secondary fluid would be circulated through the earth.

Likewise the usual method of supplying heat to spaces consists of circulating air in steady flow from the space, through the heat pump, and back to the space at some higher temperature. An alternate arrangement would be water circulating through radiators, or preferably, low temperature panels. In any event, a fluid is employed which has its temperature raised in passing through the apparatus. (One unique exception would be a panel heating system consisting of a refrigerant condensing in pipes embedded in the walls. At least one such system has been built[5].) Finally, shaft work must be supplied to the apparatus.

FIG. 2. TEMPERATURE-ENTROPY DIAGRAM SHOWING INHERENT IRREVERSIBILITY OF THE STEADY FLOW HEAT PUMP



Electric motors are usually employed for this purpose, but internal combustion engines have also been used to a limited extent[6].

Considered thus in the light of a steady flow process, (Fig. 1) the heat pump may be analyzed as a thermodynamic problem in which work is required to change air from state 3 to state 2, and the reservoir medium from state 1 to state 4.

From thermodynamics it may be seen that the minimum conceivable work required to operate an apparatus in this manner is equal to the increase in *availability* of the respective fluids passing through the apparatus, all with respect to the reservoir state.

THE AVAILABILITY CONCEPT

The heat pump together with the space it is heating may be considered as a system, operating in an environment or reservoir of infinite dimensions. The medium of this reservoir, most conveniently thought of as the surrounding atmosphere, is assumed to be in perfect equilibrium, that is, there is no opportunity for spontaneous work or heat transfer to occur within it. When the system and reservoir together, called the overall system, are in perfect equilibrium, the system is said to be in the dead state. Availability may be defined as the maximum possible work that the overall system can produce when it is in any given state.

Equations for evaluating the availability of any system, either non-flow or steady-flow, can be derived with thermodynamic reasoning based on the First

and Second Laws of Thermodynamics. It has been shown[7] that the availability, A, of a unit steady flow system is expressed by:

$$A = \left(u_1 + P_1v_1 - T_0s_1 + \frac{V_1^2}{2g} + z_1\right) - (u_0 + P_0v_0 - T_0s_0 + z_0)$$

or

$$A = \left(h_1 - T_0s_1 + \frac{V_1^2}{2g} + z_1\right) - (h_0 - T_0s_0 + z_0) \quad \ldots\ldots \quad (1)$$

where

A = availability (available energy).
u = specific internal energy (internal energy per unit mass).
p = pressure.
v = specific volume.
T = absolute temperature.
s = specific entropy.
$\dfrac{V^2}{2g}$ = specific kinetic energy.
z = specific potential energy.
h = $u + pv$, by definition (enthalpy).

All units are consistent, and numerical subscripts refer to sections across the steady flow path. The subscript $_0$ refers to the system in the dead state.

It is to be noted that all the terms of Equation 1 are properties of the system, and therefore availability is itself a composite property of a system.

It follows from Equation 1 that when a fluid changes state between section 1 and section 2 (See Fig. 1) along a path of steady flow, the increase in specific availability, ΔA, is

$$\Delta A_{1\text{-}2} = \left(h_2 - T_0s_2 + \frac{V_2^2}{2g} + z_2\right) - \left(h_1 - T_0s_1 + \frac{V_1^2}{2g} + z_1\right)^*;$$

and if the changes in kinetic and potential energy are negligible for fluids passing through the apparatus:

$$\Delta A_{1\text{-}2} \cong (h_2 - T_0s_2) - (h_1 - T_0s_1)^* \quad \ldots\ldots\ldots\ldots \quad (2)$$

For the usual case, where more than one fluid enters and leaves the apparatus, the rate increase in availability, Btu per hour, is equal to the sum of the increase in specific availability of all fluids multiplied by the respective mass rates of flow.

HEAT PUMP ANALYSIS BASED ON AVAILABILITY

The author here proposes to examine in more detail the functioning of the heat pump as a thermodynamic apparatus, the fluids circulating in steady flow, and a method of setting up criteria of performance independent of the capacity of any particular system.

It is apparent that the heat pump operates with certain variables in design conditions. These include types of fluids being circulated, the temperatures of the fluids leaving the apparatus, and the temperatures of the reservoir and the space to be heated. Standard barometric pressure is assumed for all fluids crossing the boundaries of the apparatus.

It can be shown (see Appendix A) that the rates of availability change in all fluids are identical, if the fluid is changing temperature between specified limits,

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* Numerical subscripts 1, 2, 3 and 4 refer to states (See Figs. 1, 2 and 6).

and if energy is being transferred reversibly at a specified rate. Therefore, as a matter of convenience only, air will be used as the working medium both from the reservoir and the space.

With reference to Fig. 1, air temperature at sections 1, 2 and 4 will vary widely with design and operating conditions, while the temperature at section 3 will be assumed to be the 70 F ordinarily used for inside design conditions. Specific humidity of the air, if it remains constant, has no effect on this analysis. If humidification is supplied however, the performance will be affected, but since it can be shown that its effect would be to raise the theoretical coefficient of performance, it will not be considered in this analysis.

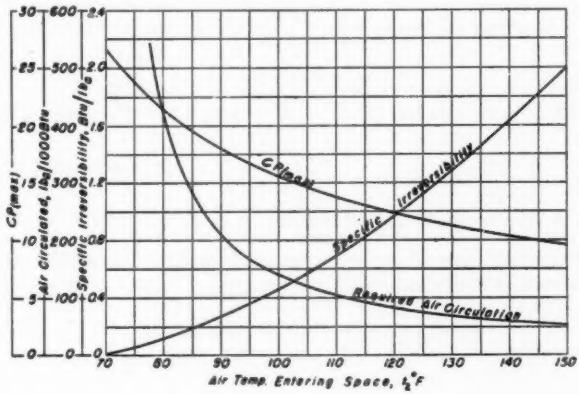

FIG. 3. PERFORMANCE OF AN IDEAL HEAT PUMP HEATING A 70 F SPACE FROM A 50 F RESERVOIR

The coefficient of performance, CP, of the steady flow heat pump (Fig. 1) is,

$$(CP) = \frac{\text{effect desired}}{\text{work input}}$$

The effect desired is a supply of energy to the space equal to the heat lost from the space:

$$Q = M_a \,(h_{m,2} - h_{m,3}) \quad \ldots \ldots \ldots \ldots \quad (3)$$

where,

Q = heat lost from the space, Btu per hour.
M_a = weight of air circulated through the space, lb_a per hour.
lb_a = pounds of dry air.
h_m = enthalpy of moist air, Btu per lb_a.

The work input will depend upon the construction of the heat pump and its operating cycle, as well as upon the temperature of the fluids entering and leaving. However, for a given set of design conditions, there is an irreducible amount of work below which a heat pump could not conceivably operate. This minimum work is equal to the rate of availability increase of all the fluids

passing through the heat pump and can be evaluated with Equation 2. This follows from the definition of availability, since all processes occurring in the overall system must be reversible for Equation 1 to be valid.

Therefore the maximum conceivable coefficient of performance $(CP)_{max}$ for the steady flow heat pump would be:

$$(CP)_{max} = \frac{M_{a,3}\,(h_{m,2} - h_{m,3})}{\Delta_2 A_2 + \Delta_1 A_4} \quad \ldots\ldots\ldots\ldots \quad (4)$$

$$= \frac{M_{a,3}(h_{m,2} - h_{m,3})}{M_{a,3}\,[(h_{m,2} - T_1 s_{m,2}) - (h_{m,3} - T_1 s_{m,3})] + M_{a,1}\,[(h_{m,4} - T_1 s_{m,4}) - (h_{m,1} - T_1 s_{m,1})]}$$

$$= \frac{(h_{m,2} - h_{m,3})}{\left[(h_{m,2} - h_{m,3}) - T_1(s_{m,2} - s_{m,3})\right] + \frac{M_{a,1}}{M_{a,3}}\left[(h_{m,4} - h_{m,1}) - T_1\,(s_{m,4} - s_{m,1})\right]} \quad (5)$$

$$= \frac{c_p\,(t_2 - t_3)}{c_p\,(t_2 - t_3) - T_1 c_p \log_e\left(\frac{T_2}{T_3}\right) + \frac{M_{a,1}}{M_{a,3}}\left[c_p\,(t_4 - t_1) - T_1\,c_p \log_e\left(\frac{T_4}{T_1}\right)\right]} \quad \ldots (6)$$

where,

c_p = specific heat of air at constant pressure, Btu per (pound) (Fahrenheit degree).

t = temperature of air, Fahrenheit.

For the limiting case where states 4 and 1 in Fig. 1 are identical and assuming that moist air behaves as a perfect gas,

$$(CP)_{max} = \frac{c_p\,(t_2 - t_3)}{c_p\,(t_2 - t_3) - T_1\,c_p \log_e\left(\frac{T_2}{T_3}\right)} \quad \ldots\ldots\ldots \quad (7)$$

INHERENT IRREVERSIBILITY IN HEAT PUMP SYSTEMS

It has been shown that the minimum work required to change the air streams (Fig. 1) from states 1 and 3 to states 4 and 2, respectively, is a measure of the increase in availability. Likewise, the decrease in availability of the air streams in changing back to their original state would be a measure of the maximum work that could be produced by the overall system. The difference between the maximum work and the actual work that results during a change of availability in a system is a quantitative measure of irreversibility. The air at state 4 is mixed with air in the reservoir, and the air at state 2 is mixed with air in the space. Obviously no work results. The process is irreversible, and consequently the irreversibility per pound of air circulated is equal to the increase in availability per pound of air circulated.

This is illustrated in Fig. 2. The crosshatched area is proportional to the irreversibility per pound of air circulated (specific irreversibility). Numerically, it is equal to the denominator of Equation 5, with the exception that area (b) should be multiplied by the ratio $M_{a,1}/M_{a,3}$ to obtain consistent units of Btu per pound of air circulated through the space.

For any assigned temperatures in Fig. 2, the irreversibilities (a) and (b) are inherent and unavoidable. The practical way of minimizing this loss is to bring t_2 and t_4 as close as possible to t_3 and t_1 respectively. There are certain limitations involved here, however. The heat loss from a building is a constant,

and as shown by Equation 3, any lowering of temperature t_2 (and consequently $h_{m,2}$) must be accompanied by an increase in the quantity of air circulated, M_a. In the limiting case when t_2 approaches t_3, M_a approaches infinity. In this case Equation 7 becomes indeterminate and can be evaluated mathematically in the following manner:

$$(CP)_{max} = \frac{(T_2 - T_3)}{(T_2 - T_3) - T_1 \log_e \left(\frac{T_2}{T_3}\right)} = \frac{0}{0} \text{ (when } T_2 = T_3)$$

$$\lim_{x \to a} \frac{f(x)}{\phi(x)} = \lim_{x \to a} \frac{f'(x)}{\phi'(x)}$$

$$\lim_{T_2 \to T_3} (CP)_{max} = \frac{f'(T_2 - T_3)}{\phi'[T_2 - T_3 - T_1(\log_e T_2) + T_1(\log_e T_3)]}$$

$$= \frac{1}{1 - \frac{T_1}{T_2}} = \frac{T_2}{T_2 - T_1} \qquad (8)$$

where T_3 and T_1 are constants

This is the Carnot Cycle coefficient of performance as applied to the heat pump, and is to be expected since all irreversibility has disappeared and the apparatus is operating between a constant temperature source and receiver.

Although it is not apparent from a casual inspection of Fig. 2, a given temperature drop $(t_1 - t_4)$ at the reservoir will always cause more irreversibility, Btu per hour, and will consequently cause a lower maximum coefficient of performance than the same temperature rise $(t_2 - t_3)$ for air entering the space. This can best be shown by examining the denominator of Equation 6. When $(t_2 - t_3) = (t_1 - t_4)$ then $\log_e T_2/T_3$ will always be smaller than $\log_e T_1/T_4$. In other words, for the case where $M_{a,1} = M_{a,2}$ and for a given temperature drop Δt_4; there is more irreversibility than for the same temperature rise Δt_2: under this condition, area (b) would be greater than area (a), Fig. 2. It can be shown that values of $M_{a,1}/M_{a,3}$ other than unity would not alter this conclusion.

This is a fortunate circumstance since practical considerations make it feasible to operate with a temperature drop $(t_1 - t_4)$ much smaller than temperature rise $(t_2 - t_3)$.

EVALUATING MAXIMUM COEFFICIENT OF PERFORMANCE

Case a. Where $t_1 = t_4$ and $t_2 \geqq t_3$ (see Fig. 1): Fig. 3 shows the limiting performance of any heat pump heating a space at 70 F from a reservoir at 50 F, for an entering air temperature range from 70 F to 150 F, and where isothermal extraction of energy from the reservoir is assumed. Such a condition could be approximated by an actual heat pump using an evaporator so as to pass reservoir fluid (air or water) over the coil in sufficiently large quantities so that the temperature drop would be negligible.

It is noted that the inherent irreversibility decreases from a maximum at $t_2 = 150$ F to zero at $t_2 = 70$ F, but that the corresponding quantity of air required to be circulated through the space increases very rapidly at values of

t_2 below 90 F to 100 F. The limiting value of t_2 in practice is usually determined by the maximum air changes possible without creating drafts.

Sample calculations for case (a).

$$\text{Design Conditions} \begin{cases} t_1 = t_4 = 50 \text{ F} \\ t_2 = 95 \text{ F} \\ t_3 = 70 \text{ F} \end{cases}$$

The specific irreversibility, from the denominator of Equation 7, equals

$$c_p\,(t_2 - t_3) - T_1\,c_p \log_e\left(\frac{T_2}{T_3}\right) = 0.241\left[(25) - 509.7 \log_e\left(\frac{554.7}{529.7}\right)\right] = 0.360 \text{ Btu/lb}$$

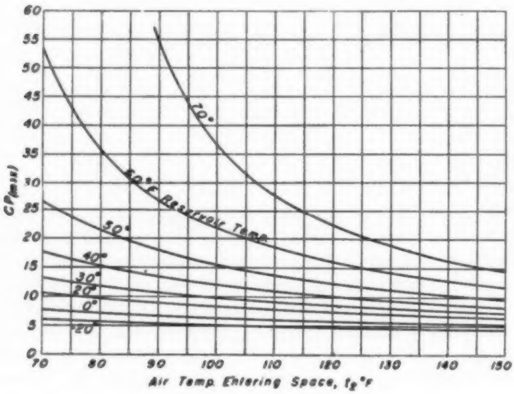

FIG. 4. $(CP)_{max}$ FOR ANY HEAT PUMP HEATING A 70 F SPACE FROM DIFFERENT TEMPERATURE RESERVOIRS (FOR THE LIMITING CASE OF ISOTHERMAL HEAT TRANSFER FROM THE RESERVOIR)

The air circulated M_a, from Equation 3, is:

$$M_a = \frac{Q}{h_{m,2} - h_{m,3}}$$

For air this becomes:

$$M_a = \frac{Q}{c_p\,(t_2 - t_3)} = \frac{1000}{0.241\ (95\text{-}70)} = 166.0 \text{ lb}_a \text{ per 1000 Btu heat lost from the space}$$

The maximum conceivable coefficient of performance $(CP)_{max}$, from Equation 7, becomes:

$$(CP)_{max} = \frac{(t_2 - t_3)}{(t_2 - t_3) - T_1 \log_e\left(\frac{T_2}{T_3}\right)} = \frac{(95\text{-}70)}{(95\text{-}70) - 509.7 \log_e\left(\frac{554.7}{529.7}\right)} = 16.73$$

For the limiting case, where $t_2 = t_3$, from Equation 8:

$$(CP)_{max} = \frac{T_2}{T_2 - T_1} = \frac{529.7}{20} = 26.48$$

Fig. 4 is supplementary to Fig. 3 in that $(CP)_{max}$ is given for additional reservoir temperatures ranging from 70 F down to −20 F.

Case b. Where $t_4 \leq t_1$ and $t_2 \geq t_3$ (see Fig. 1): This is the more general condition of heat pump operation and in this case Equation 6 can be used to evaluate $(CP)_{max}$. It is to be remembered that although Equation 6 is valid only for a perfect gas the resulting $(CP)_{max}$ will represent the limiting performance of any heat pump operating with the temperatures designated, regardless of the composition of the circulating fluids.

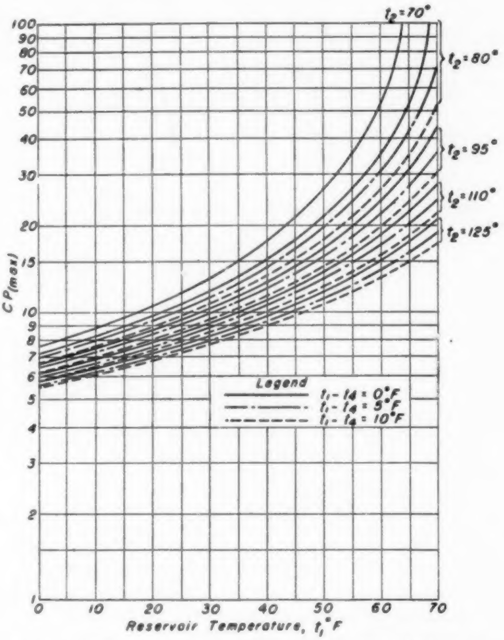

FIG. 5. $(CP)_{max}$ FOR ANY HEAT PUMP HEATING A 70 F SPACE FROM DIFFERENT TEMPERATURE RESERVOIRS (THE TEMPERATURE t REFERS TO FIG. 1)

However, before Equation 6 can be used, it is necessary to evaluate the unknown, $(M_{a,1}/M_{a,3})$ from the known temperatures in the following manner:

$$\text{Let } Q_{(in)} = M_{a,1}\, c_p\, (t_1 - t_4); \text{ and } Q_{(out)} = M_{a,3}\, c_p\, (t_2 - t_3);$$

where

Q_{in} = heat transferred to the heat pump.
Q_{out} = heat transferred from the heat pump.

In the derivation of Equation 1 it is shown that the maximum work done by any system in changing to the dead state will be that done by a reversible engine. Similarly, the minimum work required to raise the availability of a fluid would be that supplied by a reversible heat pump. Consequently the *rate* of

entropy decrease, Btu per Fahrenheit degree, for the fluid flowing between states 1 and 4 is equal to the *rate* of entropy increase, Btu per Fahrenheit degree, for the fluid flowing between states 3 and 2. This must be true since no other heat crosses the boundaries of the heat pump and all processes are assumed to be reversible. Symbolically:

$$-\Delta_1 S_4 = \int_4^1 \frac{dQ_{(in)}}{T} = \int_4^1 \frac{M_{a,1}\, c_p dt}{T}$$

$$\Delta_3 S_2 = \int_3^2 \frac{dQ_{(out)}}{T} = \int_3^2 \frac{M_{a,3}\, c_p dt}{T}$$

and therefore:

$$M_{a,1}\, c_p \log_e\left(\frac{T_1}{T_4}\right) = M_{a,3}\, c_p \log_e\left(\frac{T_2}{T_3}\right)$$

or

$$\frac{M_{a,1}}{M_{a,3}} = \frac{\log_e (T_2/T_3)}{\log_e (T_1/T_4)} \quad \ldots\ldots\ldots\ldots \quad (9)$$

substituting (9) in Equation 6

$$(CP)_{max} =$$

$$\frac{c_p (t_2 - t_3)}{\left[c_p (t_2 - t_3) - T_1 c_p \log_e\left(\frac{T_2}{T_3}\right)\right] + \frac{\log_e (T_2/T_3)}{\log_e (T_1/T_4)}\left[c_p (t_4 - t_1) - T_1 c_p \log_e\left(\frac{T_4}{T_1}\right)\right]}$$

$$= \frac{(t_2 - t_3)}{(t_2 - t_3) - \frac{\log_e (T_2/T_3)}{\log_e (T_1/T_4)} \times (t_1 - t_4)} \quad \ldots\ldots\ldots\ldots \quad (10)$$

Fig. 5 shows the $(CP)_{max}$ for any heat pump heating a space at 70 F from any reservoir down to 0 F. Four entering air temperatures, $t_2 = 80$, 95, 110, and 125 F, are included, and for each one of these conditions, performances are shown for three temperature drops to the reservoir, $(t_1 - t_4) = 0$, 5, and 10 F. A curve showing the $(CP)_{max}$ for the limiting case, $t_1 = t_4$ and $t_3 = t_2$, is shown for comparative purposes. Values for $(CP)_{max}$ for design conditions other than those shown can be estimated by interpolation.

Sample calculations for case (b).

$$\text{Design conditions} \begin{cases} t_1 = 50\text{ F} & t_4 = 40\text{ F} \\ t_3 = 70\text{ F} & t_2 = 95\text{ F} \end{cases}$$

From Equation 10,

$$(CP)_{max} = \frac{(95\text{-}70)}{(95\text{-}70) - \frac{\log_e (554.7/529.7)}{\log_e (509.7/499.7)} \times (50\text{-}40)} = 14.53$$

SUMMARY AND CONCLUSIONS

Experience has shown that the practical way to design and operate a heat pump is as a steady flow apparatus in which fluids enter and leave the equipment with falling or rising temperatures. The amount of temperature change

in these fluids will depend on their mass rates of flow which are limited by factors such as permissible air changes, friction drop, noise, circulating fan and pump horsepower.

The minimum conceivable work required to change the state of these fluids between specified limits can be evaluated by calculating their change in availability.

Thus, a relatively simple method can be used for evaluating the performance for the various ideal and operating heat pump cycles of common engineering installation. Such an evaluation will show that the ideal simple vapor compression cycle falls far short of the maximum theoretical coefficient of performance. Various complex vapor compression cycles show considerable improvement in this regard, and the maximum possible coefficient of performance can be attained by certain ideal gas (air) cycles. A study of a number of such cycles has been made, and it is proposed to publish these results at a later date.

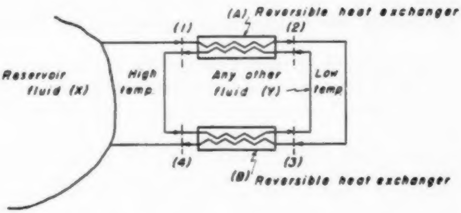

FIG. 6. RATE OF AVAILABILITY CHANGE BETWEEN TWO FLUIDS

APPENDIX A

RATES OF AVAILABILITY CHANGE IN FLUIDS

The rates of availability change in all fluids having a constant specific heat are identical, if the fluid is changing temperature between specified limits, and energy is being transferred at a specified rate. This can be demonstrated by reference to Fig. 6.

Fluid (X) has its availability decreased in passing through reversible heat exchanger (A) from high temperature state 1 to low temperature state 2. However, fluid (Y) has its availability increased in passing from state 2 to 1. Inspection shows that these two changes in availability are occurring at equal rates (but not necessarily equal per pound of fluid circulated) since another reversible heat exchanger (B) could be employed to change the availability of fluid (X) from state 3 back to the reservoir value. But fluid (Y) is *any* fluid of constant specific heat and therefore the product of the mass rate of flow, pounds per hour, and the change in availability, Btu per pound, is a constant irrespective of the fluid.

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DISCUSSION

F. EDWARD INCE, St. Louis, Mo.: The author mentioned a maximum discharge temperature of 95 F, yet some of us know of instances where it is a definite problem even with discharge temperatures of 110 F. On the other hand, we know that the efficiency of the pump falls off very rapidly with the higher compression temperatures.

I would like to have a brief explanation concerning the irreversibility of the throttle valve.

Although it is convenient to calculate and use the temperature in calculations as correlated with the work done; it is not too directly proportional, as undoubtedly the gas volume in the vapor compression cycle does not follow the direct function of the temperature.

When we speak of air cycle efficiencies of 88 percent and of other air cycles as 100 percent, I am wondering about the overall coefficient of performance of these types of systems since the work efficiency of the air cycle for a horsepower input is very low.

AUTHOR'S CLOSURE: The inference that Mr. Ince made that 95 F entering air may result in drafts is well taken. For instance a 70 F space with 95 F entering air will have 8.9 air changes per hour when the heat loss is only 4 Btu per cu ft. However this low temperature was deliberately chosen for the illustrative problems to dramatize the need for lowest practical entering air temperatures to take advantage of maximum coefficients of performance. Any thing we can do to minimize the heat loss from the space will aid in reducing the required temperature of air entering the space in relation to the acceptable number of air changes.

The point was made that the efficiency of the heat pump falls off very rapidly with higher compression temperatures. This is particularly true of the simple vapor compression cycle. It is interesting to note that the maximum conceivable coefficient of performance as calculated by Equation 10 does not fall off nearly as rapidly at higher discharge temperatures and that therefore the inherent limitations of the simple vapor compression cycle become more pronounced at these temperatures.

The simplest way to explain irreversibility of the throttling process is to consider it as a loss. There is a loss in available energy due to throttling the refrigerant from some higher pressure to some lower pressure. In theory it would be possible to recover this energy, with a fluid turbine for instance, but of course this is not attempted from practical reasons in the vapor compression cycle.

An analysis of ideal air cycles will show that some of them can equal the maximum conceivable coefficient of performance when employed as a steady flow heat pump as shown in Fig. 1. The fact that they were abandoned many years ago by the refrigeration industry does not alter this fact and perhaps a re-examination of these cycles in the light of more recent developments in air handling equipment is indicated.

1382

No. 1382

EVALUATING HEAT PUMP PERFORMANCE

By F. R. Ellenberger*, A. B. Hubbard**, W. R. Foote**, F. Burggraf**, J. J. Martin, Jr.**, Bloomfield, N. J.

Any heat pump system consists of a series of components—evaporator, motor, compressor, condenser, liquid throttling device, auxiliaries, etc., each of which has certain inherent operating characteristics. To integrate these individual characteristics under variable conditions of heat source and heating requirements seems a formidable analytic procedure. Consequently, the expedient of arbitrarily assumed *design* conditions is usually resorted to in predicting performance. This prevailing tendency for over-simplification of heat pump performance is also due to scarcity of significant operating data.

It is becoming increasingly apparent that sizing methods prevalent for combustion heating are not adequate for heat pump heating, but that more precise data must be obtained before it will be possible to determine the best heat pump design, considering the effect of climate, the nature of the heat source and the influence of the building structure[1].

An analytical method is presented herein for more exactly predicting overall heat pump performance for heating operation. It includes consideration of heat source, temperature variations and climate. A method is also presented for measuring actual performance both to check calculations and to supplement them with actual operating data not amenable to analysis.

General Design Principles

Before presenting the method developed by the authors for analyzing heat pump performance, it should be in order to briefly discuss first some basic design principles that must be observed in initially sizing components, particularly for heat pumps which obtain heat from the outdoor air.

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Presented at the 56th Annual Meeting of The American Society of Heating and Ventilating Engineers, Dallas, Tex., January 1950.

[1] Exponent numerals refer to Bibliography.

In designing a heat pump the first consideration is the choice of a compressor and its driving motor. Basic capacity characteristics will dictate the initial choice, and the problem is then to determine how much heating and cooling can be obtained with the combination chosen.

Studies show that the compressor usually requires its maximum power during cooling operation at high outdoor temperatures, and the compressor speed is chosen to restrict the motor power under those conditions.

Anomalies may arise, for example, if multiple compressor units are to be used for heating. Here the speed of one compressor unit is determined to limit power under cooling conditions; successive compressor units are chosen to limit power

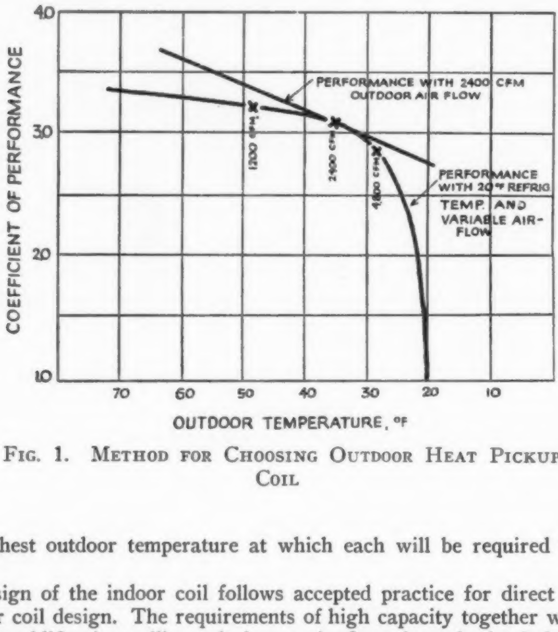


FIG. 1. METHOD FOR CHOOSING OUTDOOR HEAT PICKUP COIL

at the highest outdoor temperature at which each will be required to furnish heat.

The design of the indoor coil follows accepted practice for direct expansion evaporator coil design. The requirements of high capacity together with appropriate dehumidification will result in an air flow through the indoor coil of approximately 400 cfm per ton of cooling. Such a coil is then generally satisfactory for heating on the reverse cycle as a condenser, if proper attention is given to refrigerant drainage.

Selecting the heat pick-up coil provides the designer with the widest range of choice. Fig. 1 illustrates one method that has been found helpful for an air-source heat pump. This curve indicates that there is a coil size that results in an optimum coefficient of performance (CP). The straight line shows the relation of CP with outdoor temperature with a fixed system. The curved line shows the change of CP if the suction pressure is held constant and the outdoor coil and air flow are varied. An optimum is determined where the two lines are tangent.

Physically, the interpretation is quite simple. A very large outdoor coil and corresponding air flow require so much fan power that the CP is reduced. A very small coil and air flow will operate at a low suction pressure, and the CP is accordingly reduced. Between these two extremes there is an optimum that will provide the highest CP.

This optimum coil will be subject to less hours of frosting per season, and during those hours the rate of frosting will be less. Being generously sized, its surface temperature will be close to that of the entering air, and as a consequence, the proportion of heat removed from the air, by condensation of water vapor, to sensible heat removal, will be small. Also a high surface temperature will permit operation for longer periods above 32 F than would a smaller coil with a lower surface temperature for the same entering air temperature.

A properly sized outdoor coil will, in addition, be large enough to provide for permissible condensing pressure for cooling operation.

With the major components of a heat pump system sized in accordance with the foregoing considerations, it is now desirable to check expected operation by more detailed means.

COMPONENTS MEET CHANGES IN OPERATING CONDITIONS

Each component of any machine has an operating characteristic which can usually be put in graphical form. The horizontal scale of such a plot represents the imposed condition, known as the independent variable. The relationships between points on the horizontal scale to those on the vertical scale are indicated by one or more lines or curves, each labeled with the values of other imposed conditions. These are called parameters. The numbers read off on the vertical scale portray a dependent variable, *i.e.,* the operating result obtained when the machine is required to meet a combination of two or more imposed operating conditions.

A direct-current motor affords a convenient example. The connected machine, in response to the requirements imposed on it, fixes the torque, or turning effort, at the motor shaft. A given voltage will then result in a definite motor speed. In this case the torque, voltage and speed are, respectively; independent variable, parameter and dependent variable.

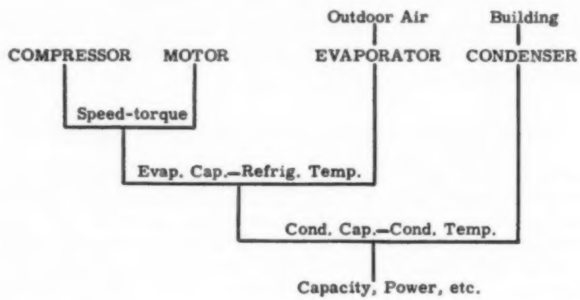
The example could be carried further by considering the characteristic of a machine driven by the motor. If this were a pump, for example, its operation could also be plotted on the same speed and torque scales but the parameter lines might be labeled as water flows at specified pressures. The pump lines cross those of the motor, revealing new relations with the result that speed and torque no longer need be considered. In other words, the operating characteristic of the motor and pump as a unit now consists of voltage, water flow and pressure; any one of which might be an imposed condition, with one of the remainder as a controllable variable, while the third is a desired result.

The major subassemblies of an air-to-air heat pump have interlocking characteristics, meaning that a change of any imposed condition sets up a series of automatic adjustments in the operation of the components. The wide range of temperature and moisture combinations of the outdoor air heat source must be accounted for. A range of heating and cooling loads must be supplied as dictated by the combination of building structure, climate, and usage.

Variations in electric power supply and water temperature (for auxiliary heat source) must be considered. From the many combinations and proportions available, the heat pump designer must select those which will lead to good value from the user's viewpoint—the most and best for the least. Finally, the heating or cooling must be delivered to the conditioned space in a way acceptable to the user. He does not want, for example, a gale of 22,500 cfm of cold 72 F air, nor only 360 cfm at a hot 200 F, but perhaps, 1500 cfm at 100 F.

It would be prohibitively expensive to develop an air-to-air heat pump by trying out dozens of combinations in full-range laboratory or field tests. Instead, the designer must make a whole-hearted attempt at a complete analysis on paper rather than the customary two or three calculated points to be connected by experience and intuition.

TABLE 1—COMPONENT CHARACTERISTICS



The heat pump circuit can be linked by component characteristics as diagrammed in Table 1.

A specific procedure for combining the component characteristics is given, starting with the compressor motor and assuming the unit on heating duty:

Compressor Motor. Fig. 2-A is a speed torque curve based on compressor speed and compressor torque (by putting in ratio of compressor to motor pulleys as one set of parameters.) For convenience, percent-full-load curves are also plotted.

Compressor (Speed-Torque). Fig. 2-B is the compressor power characteristic plotted on the same speed and torque coordinates used in Fig. 2-A for the motor. Suction saturated refrigerant temperature is the main parameter. (This is equivalent to suction pressure or suction refrigerant density, but is a better common denominator as will be seen later). The condensing temperature (synonymous with compressor discharge pressure but again more convenient here) is not yet decided for this system but has to be known in order to calculate Fig. 2-B. To get around this difficulty, three transparent curves are made; one each for 110 F, 120 F, and 130 F condensing temperatures in the example being cited. By advance inspection it was known that the balancing temperatures will be near this range.

Motor-Compressor Unit. Fig. 2-A is superimposed on Fig. 2-B, as in Fig. 2-C, to obtain a combined characteristic for the motor and compressor. The three charts in Fig. 2-B can be used in turn as overlays to give compressor *operating lines* which

relate suction saturated temperature, speed, and torque for any desired pulley ratio. Compressor shaft horsepower can be calculated in turn from speed and torque. Each series of performance figures is for the condensing temperature noted on the Fig. 2-B overlay used.

Compressor Cooling Capacity. By assuming no superheat at compressor inlet and no subcooling of refrigerant liquid, compressor cooling capacity (heat pick-up) and

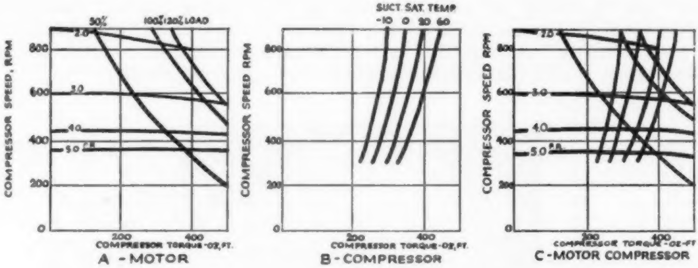


FIG. 2. GRAPH FOR HEAT PUMP PERFORMANCE ANALYSIS

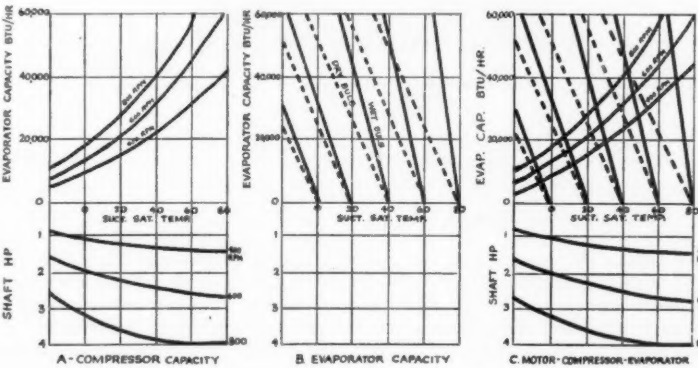


FIG. 3. GRAPH FOR HEAT PUMP PERFORMANCE ANALYSIS

compressor shaft horsepower can be plotted against suction saturated temperature with compressor speed as a parameter. For this purpose, the two-quadrant curve in Fig. 3-A is convenient. Again, three transparent charts for condensing temperatures of 110, 120 and 130 F are used.

Evaporator Cooling Capacity. In the case of an evaporator, cooling capacity (heat pick-up) can be plotted against suction saturated refrigerant temperature with the dry bulb and wet bulb temperatures of the entering air as parameters. Fig. 3-B is a transparent chart with scales corresponding to the upper quadrant of Fig. 3-A. A different chart must be available for each evaporator air flow contemplated. Space

does not permit describing in detail how this chart was derived. However, Fig. 5 indicates the method by which evaporator coil heat transfer performance can be summarized on a small transparency which serves as a rapid calculator when placed over an appropriate psychrometric chart.

Motor-Compressor-Evaporator Unit. Fig. 3-B is placed over the upper quadrant of Fig. 3-A to form the chart shown in Fig. 3-C. Condensing temperature, against which the combination works, is determined by the choice of Chart 3-A; while the evaporator air flow choice depends on Chart 3-B. The motor and pulley ratio choices are transferred from Fig. 2-C by plotting compressor *operating lines* on Fig. 3-C (taking care that the condensing temperatures of Figs. 2-B, 2-C, 3-A, and 3-C correspond). At this point it can be mentioned that the overlaying transparencies, complete with inscribed curves, can be blueprinted to preserve a record of the results.

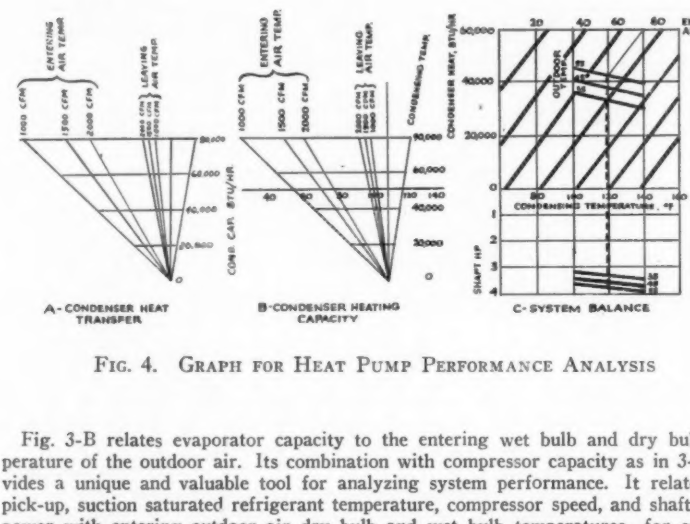


FIG. 4. GRAPH FOR HEAT PUMP PERFORMANCE ANALYSIS

Fig. 3-B relates evaporator capacity to the entering wet bulb and dry bulb temperature of the outdoor air. Its combination with compressor capacity as in 3-C provides a unique and valuable tool for analyzing system performance. It relates heat pick-up, suction saturated refrigerant temperature, compressor speed, and shaft horsepower with entering outdoor air dry bulb and wet bulb temperatures—for specified compressor motor and drive, outdoor coil air flow, and condensing temperature.

After establishing a compressor *operating line* on Fig. 3-C, the compressor-evaporator balance point can be found for any particular combination of wet and dry bulb temperature.

If the temperature point falls below the operating line, the coil will operate wet (or frosted) and wet bulb temperature *rules*. That is, one should follow up the slanting wet bulb line to an intersection with the operating line to obtain refrigerant temperature, cooling capacity, compressor speed, and go down the refrigerant line to the lower curve to obtain horsepower. If the temperature point falls above the operating line, the coil is operating dry and dry bulb temperature *rules*. Follow down the slanting dry bulb line to an intersection with the operating line to obtain the balance point.

System Balance. The condenser remains as the last component to be considered in order to close the heat pump circuit. Heretofore, three separate condensing temperatures of 110, 120 and 130 F have been carried through the calculations. Now, how-

ever, it is necessary to establish a specific condensing temperature for each condition of operation in order to complete the system balance. For this purpose, condenser heat output is determined from the motor-compressor-evaporator combination and is cross plotted with condenser heat transfer characteristics to establish a condensing temperature.

Fig. 4-A shows a convenient method of summarizing condenser heat transfer performance. When superimposed over a temperature scale, as in Fig. 4-B, the condensing temperature is indicated at the intersection of the T_c line with the horizontal temperature scale for any desired air flow. Capacity, entering air temperature and leaving air temperature can be read off for any combination. Data obtained from 4-B are then used to construct the slanting-line chart in the upper quadrant of Fig. 4-C. This is condenser heating capacity plotted against condensing temperature with entering air temperature as the parameter. The near-horizontal lines in the upper quadrant are condenser heating capacities as approximated by adding evaporator capacity and compressor input (in Btu per hour). The intersection of two lines for any appropriate set of conditions indicates the condenser balance point. Condenser capacity and condensing temperature can now be read off.

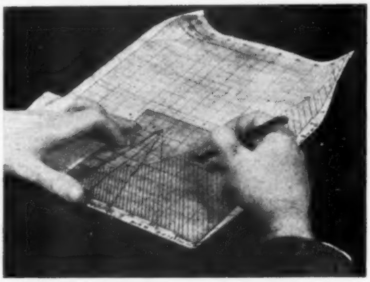


FIG. 5. ILLUSTRATING EVAPORATOR PERFORMANCE ANALYSIS

The lower quadrant of Fig. 4-C relates compressor input with condensing temperature for various outdoor air conditions as obtained from Fig. 3-C. It is now possible to establish evaporator capacity with the system balance thus obtained.

Net Heating Capacity. The net heat pump heating capacity can be obtained by the following relation:

$$Q_h = Q_{ev} + Q_{cm} + Q_{fm}$$

where

Q_h = Net heat output.
Q_{ev} = Evaporator capacity.
Q_{cm} = Thermal equivalent of compressor motor input.
Q_{fm} = Thermal equivalent of fan power that is recoverable as useful heat.

USE OF PERFORMANCE CURVES

The precise method by which the performance curves of Figs. 2 to 4 are used depends upon the exact heat pump system contemplated. In the case of a heat

pump, for example, which obtains heat from a source other than outdoor air, curves of a different nature will be required for Fig. 3-B.

Fig. 6 illustrates the overall heating performance of a specific heat pump, obtained by applying the described calculation methods. Heating capacity, sup-

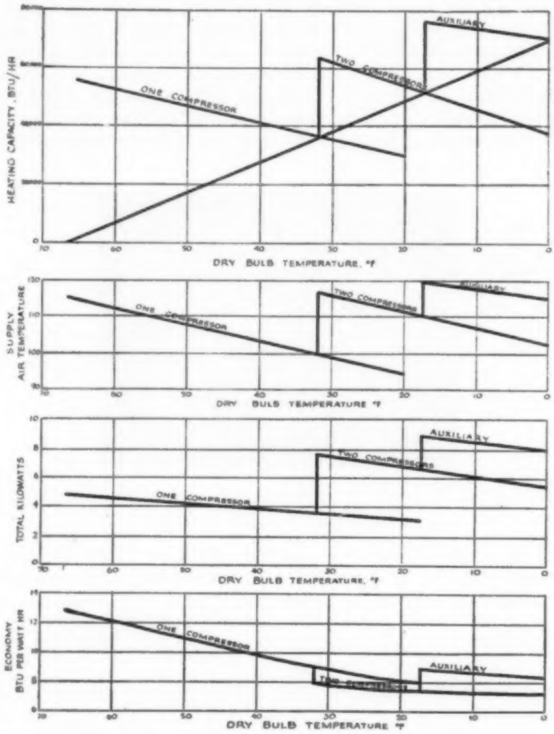


FIG. 6. HEAT PUMP PERFORMANCE CURVES FOR HEATING

ply air temperature, electrical input and economy are all related to outdoor temperature. The heat pump cited has the following general specifications:

1. Heat pump uses outdoor air as basic heat source and water as an auxiliary heat source.
2. Two compressors are incorporated in the same refrigeration circuit, only one of which is used for cooling operation, while either one or both is used for heating, depending on the demand for heat.

To make the fullest possible use of heat pump performance characteristics, they must be extended to take into account the building to be air conditioned and the climate to be experienced.

EFFECT OF BUILDING AND CLIMATE

Two climate factors affect heating performance: *first,* the range in dry bulb temperature, and *second,* the range in humidity or wet bulb temperature. Another very important climate factor is the minimum expected temperature, because

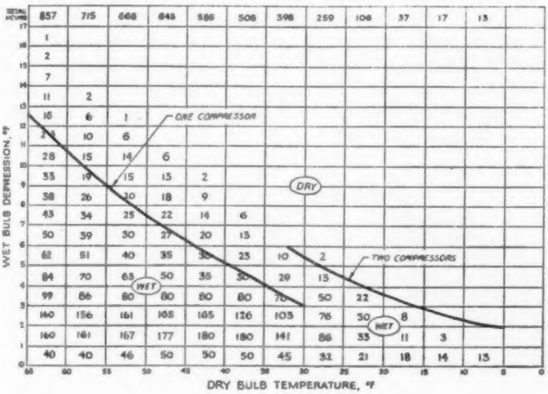


FIG. 7. WEATHER CONDITIONS INFLUENCING HEAT PUMP OPERATIONS

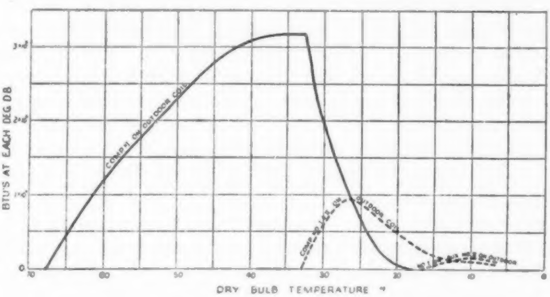


FIG. 8. TOTAL HEAT PRODUCED FOR TYPICAL SEASON

this will determine, for a particular heat pump, the largest building which can be satisfied in that climate. To a lesser degree, the humidity factor of the climate will affect the performance of the unit, and in addition, determine the amount of outdoor coil defrosting necessary.

The climate can be most readily presented for calculating purposes as an *hours at* tabulation of wet bulb depression *vs.* dry bulb temperature as on Fig. 7. This particular climate is that of Chattanooga, Tenn.[2] It is interesting to note

how the hours tend toward the zero depression line (100 percent humidity). It can be noted that most of the operating hours are in the *wet* region of unit operation. In Fig. 7, the outdoor dry bulb temperature is in 5 degree steps; this simplifies the figure and also shows that climatic conditions are fairly regular on a yearly basis, even though daily changes often can be very rapid.

Let us now consider heat pump performance on heating under these foregoing conditions. Of course, at a particular dry bulb temperature several capacities are encountered depending on the wet bulb depression and ranging from maximum at saturation to minimum at dry coil operation. Calculation has shown that this variation in operation is essentially linear with wet bulb depression. Hence with the *hours at* data given in Fig. 7, and the maximum and minimum capacities at a particular dry bulb, a median capacity can be

TABLE 2—HEATING NEEDS

TEMPERATURE RANGE	HEATING NEEDED Btu (millions)	POWER REQ'D kwhr	COMPRESSOR OPERATION	HRS
32 to 70	74.9	7,220	No. 1 alone	2146
15 to 32	21.1	2,810	No. 1 and No. 2	195
5 to 15	1.9	250	Auxiliary	15
	97.9	10,280		

calculated for each particular dry bulb temperature. It is to be noted that the capacity curves in Fig. 6 only show this weighted average curve. Another climate may present a slightly different humidity pattern.

The combination of the performance curves with the *hours at* climate data can yield a good picture of the heat pump operation. For instance, for this climate and building, 97.9 million Btu are needed during a heating season while using 10,280 kwhr. Table 2 shows how these heating needs are split as a function of outdoor temperature. Less than 2 percent of the heating load is needed between 5 and 15 F. Thus the typical operation can be seen where the weather has a long *tail* covering a number of degrees but involving very few hours. A pictured representation presented elsewhere illustrates this[3].

Another picture of the typical heat pump operation is obtained by plotting the Btu delivered at a particular dry bulb over the heating season as in Fig. 8. Here it can be seen that the No. 1 compressor alone takes the major load. The hours of operation of each combination can be calculated as listed in Table 2. This shows the great usefulness of the auxiliary heat source. Essentially there are very few hours between 5 and 15 F in this typical climate, but these hours nevertheless must be taken care of by the heat pump. To keep down the equipment investment some auxiliary source of heat is very desirable. If water were used at the rate of 10 gpm for the auxiliary hours, this would amount to less than $10.00 a year at nominal rates. This is worthwhile to conserve on equipment size.

These are, of course, but a few of the many pieces of information which can be gathered from the calculated combination of building and climate. The discussion of the combination has been centered about the maximum building size

that can be heated by the given heat pump since it is the most critical case. As building size decreases, obviously less auxiliary heat is necessary for this heat pump to operate satisfactorily. For a still smaller building (or the same building in a milder climate), no auxiliary heat would be necessary at all. At some point economy would dictate the need for a smaller heat pump model.

FURTHER WORK NECESSARY

Some points in the previous discussion could be elaborated further. For instance, it is well known that there is a lag between weather and its effect on the inside of a building. In the point-by-point calculation it was assumed that

FIG. 9. APPLIANCE STORE USED AS "GUINEA PIG" FOR HEAT PUMP

the building heat loss was a function only of the outdoor temperature and that the heat pump would be obtaining heat from outdoor air at that temperature. Actually, the time lag of the building tends to reduce the maximum amount of heat required so that the calculated size is conservative. Also, the assumption that the heat pump must have sufficient capacity to take care of the minimum expected temperature is conservative, since generally these minimums are of short duration and occur at night when comfort requirements are not usually so strict.

The cycling operation of the heat pump cannot be completely described from these steady-state calculations. Further analysis could attempt to appraise this type of operation but testing in the field appears to be the best way to get an exact answer.

In making these calculations no allowance was made for any ventilation air. A ventilation schedule could be set up with very little sacrifice in economy in return for many comfort benefits.

Analytical results such as these often lead to valuable features and suggest refinements of heat pump design. Their basic worth, however, is to serve as

a tool in calculating heat pump performance in specified climates so that the proper kind and size of machine can be installed and give comfortable air conditions the year 'round.

FIELD TESTING PROGRAM

As a means for obtaining reliable operating data under actual conditions of use, a field testing program is under way. Packaged heat pump units have been installed in different types of buildings in widely different climates. Most

FIG. 10. PACKAGED HEAT PUMP IN STORE

of these units have been installed by various electric utility companies which are cooperating in the testing program.

One of the installations is in an appliance store in Bloomfield, N. J., shown in Fig. 9. This heat pump is basically an air-to-air unit, the operating scheme of which was previously outlined. One of its two compressors operates during cooling, and one or both compressors can be used for heating depending on capacity requirements. There is also provision for augmenting the air source by obtaining heat from water when additional capacity is needed during cold weather peaks.

The store is a rectangular brick building with an area of 1300 sq ft and contains no internal partitions. The front two-thirds of the store is used for displaying merchandise and is divided by a counter from the manager's office. The heat loss, calculated by standard methods, is 90,145 Btu per hr and the heat gain 42,244 Btu per hr.

The heat pump is located in the front section of the store, near a side wall. Fig. 10 shows the unit located within the store. The outdoor ducts are attached to the back of the unit and pass through the south wall to the outside, where

they are separated in order to minimize recirculation between the intake and discharge. The conditioned indoor air is discharged through a grille across the front of the unit. Adjustable vanes are set for even air distribution throughout the store. Conditioned air to be recirculated is drawn in through filters at the back of the unit. The amount of outdoor air is regulated by an adjustable damper within the unit. A wall-mounted thermostat controls the temperature in the store in response to a single setting. The heat pump either heats or cools the store in order to maintain the required comfort conditions.

The Joint Heat Pump Committee of the *Association of Edison Illuminating Companies* and the *Edison Electric Institute* has outlined a standardized field test method for determining the coefficient of performance and performance fac-

FIG. 11. SPECIAL TEST PANEL FOR HEAT PUMP FIELD TESTS

tor for heat pump installations.[4] Definitions adopted by the Joint *A.E.I.C.-E.E.I.* Heat Pump Committee are as follows:

The heating coefficient of performance of an installed heat pump shall be the total useful heating effect produced by the heat pump system at stated conditions, divided by the heat equivalent of the total energy input to the system.

The term *performance factor* shall be employed when referring to a value based on an extended period of time, stating the period of time covered.

The Joint *A.E.I.C.-E.E.I.* Heat Pump Committee has suggested that the measurements taken in field tests should be such as to provide actual heat pump output based on known characteristics of the compressor and refrigerant (*calibrated compressor*). Measurements of air quantity in the field can seldom be made with acceptable accuracy, and consequently air flow-temperature methods of measuring output are not acceptable for field tests.

HOUR	MIN.	MIN. ELAPSED	STEP NO.	KW-HR. HEAT PUMP	KW-HR. DEFROST HEATER	KW-HR. AUX. RES.	KW-HR. THIS HR. DEFROST HEATER	KW-HR. THIS HR. STEP 1	KW-HR. THIS HR. STEP 2	KW-HR. THIS HR. STEP 3	KW-HR. THIS HR. STEP 4	KW-HR. THIS HR. STEP 5	KW-HR. THIS HR. STEP 6	KW-HR. THIS HR. STEP 7	HEAD PRESS.	BACK PRESS.	OUTSIDE D.B.	OUTSIDE D.P.	INSIDE D.B.	INSIDE D.P.	DISCH. TEMP	WATER TEMP IN	WATER TEMP OUT	WATER TEMP ΔT	B.T.U. STEP 1	B.T.U. STEP 3	B.T.U. STEP 4	B.T.U. STEP 5	B.T.U. STEP 6	REMARKS
3195	04	04	2	200.7											82	43					74									
	32	28	3	202.3						1.6					130	22					94					15,540				
	43	11	2	202.4					0.1						85	40					71									
3196	—	17	3	203.3						0.9					130	20	41	38	70	44	94					8,835				
									0.1	2.5							41	38	70	44						23,875				Hourly Summary
3196	13	13	3	204.1						0.8					130	20					94					6,115				
	23	10	2	204.2					0.1						85	38					71									
	55	32	3	206.0						1.8					130	20					94					17,760				
3197	—	5	2	206.0											80	30	40	38	70	44	75									
									0.1	2.6							40	38	70	44						23,875				Hourly Summary
3197	05	5	2	206.1					0.1						85	38					70									
	37	32	3	208.0						1.9					125	22					94					17,920				
	48	11	2	208.0											90	42					71									
3198	—	12	3	208.6						0.6					130	22	42	38	70	44	92					6,300				
									0.1	2.5							42	38	70	44						24,220				Hourly Summary
3198	11	11	3	209.2						0.6					140	20					92					1,320				
	24	13	2	209.3					0.1						90	45	45				71									
	40	16	3	210.3						1.0					140	22	49	40			94					10,800				
	55	15	2	210.3											90	45	50				71									
3199	—	5	3	210.6						0.3					130	22	51	42	70	44	90					2,775				
									0.1	1.9							49	40	70	44						14,895				Hourly Summary
3199	08	8	3	211.1						0.5					120	22	50				94					4,520				
	25	17	2	211.2					0.1						90	45	57	45			71									
	37	12	3	211.9						0.7					140	25	58				94					7,020				
	58	21	2	212.0					0.1						90	48	60				71									
3200	—	2	3	212.2						0.2					120	26	60	44	70	44	85					1,240				
									0.2	1.4							58	44	70	44						12,780				Hourly Summary
3200	08	8	3	212.7						0.5					140	22	62				95					4,320				
	34	26	2	212.8					0.1						90	50	65				71									
	42	8	3	213.3						0.5					120	25	66				95					4,880				
3201	—	18	2	219.4					0.1						85	50	68	44	70	44	71									
									0.2	1.0							65	44	70	44						9,200				Hourly Summary
3201	14	14	2	219.6					0.2						90	22	69				71									
	22	8	3	214.0						0.4					125	28	70				95					4,900				
3202	—	38	2	214.2					0.2						80	58	71	46	70	45	71									
									0.4	0.4							70	46	70	45						4,900				Hourly Summary

FIG. 12. SAMPLE HEAT PUMP LOG SHEET

Since the heat pump for the installation described herein was completely fabricated in the factory, it was practicable to test it in the laboratory with accurate measuring equipment before installation in order to obtain exact calibrations of its performance. Readings were thus obtained of its heating and cooling capacity over a complete range of operating conditions for each mode of operation, from which curves were prepared for gaging performance after installation.

Instruments for indicating the required field test data are mounted on a panel board in the basement of the store. A 16 mm movie camera, arranged for taking

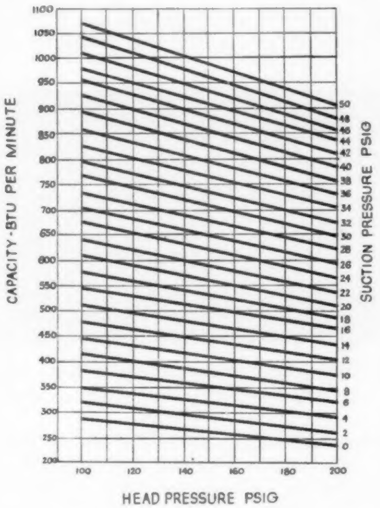


FIG. 13. SAMPLE CALIBRATION CURVE, HEATING CAPACITY WITH ONE COMPRESSOR OPERATING

single frame exposures, is focused on the panel board, and relays connected to the heat pump controls cause the camera to take one picture each time the heat pump starts or stops any of its possible seven modes of operation. In addition to these pictures, an exposure is made of the panel board every hour, on the hour, regardless of heat pump operation. The 2000 frames in one magazine of film will last from two to four weeks depending on the cycling of the machine, which in turn is dependent on the weather. Four reflector flood bulbs illuminate the board during each exposure.

Fig. 11 shows the recording panel. The information obtained from the instruments on the panel board consist of: *a.* hour of year since Jan. 1, 00 a.m. Standard Time; *b.* minute (0 to 60); *c.* kilowatthour reading of heat pump power; *d.* kilowatthour reading of power for defrost storage heater; *e.* mode of operation; *f.* outdoor dry bulb temperature; *g.* outdoor dew point temperature; *h.*

indoor dry bulb temperature; *i.* indoor dew point temperature; *j.* indoor air supply temperature; *k.* auxiliary water temperature *in; l.* auxiliary water temperature *out; m.* compressor suction pressure; *n.* compressor discharge pressure; *o.* indication of thermostat setting; *p.* any remarks that have been noted in chalk on the panel board.

DATA ANALYSES

The previous section has described the film data that are taken. This section deals with the method of utilizing the data recorded on the film.

Fig. 12 shows a log sheet upon which the data are transcribed. A film reader is conveniently used to enlarge and project each frame on a ground glass screen.

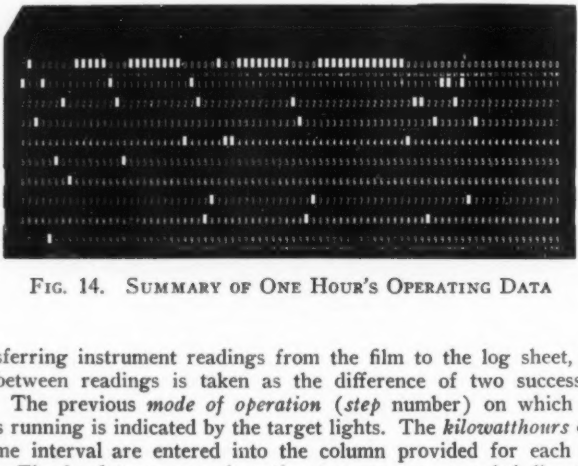


FIG. 14. SUMMARY OF ONE HOUR'S OPERATING DATA

For transferring instrument readings from the film to the log sheet, the *time interval* between readings is taken as the difference of two successive time readings. The previous *mode of operation* (*step* number) on which the heat pump was running is indicated by the target lights. The *kilowatthours* expended in the time interval are entered into the column provided for each mode of operation. The *head pressure* and *suction pressure* are recorded directly. *Temperatures* are recorded where there is a detectable change from preceding readings.

The treatment of the Btu per hour capacity of the system needs some explanation. In accordance with the *A.E.I.C.-E.E.I.* code, the heat pump can be calibrated in terms of the discharge and suction pressures and temperatures. It has been the author's experience that a liquid-suction heat exchanger in the system will maintain the suction temperature so nearly constant that this variable may be neglected. Hence a calibration may be presented, as in Fig. 13, for one mode of operation, for total capacity related to discharge and suction pressure. Similar figures are made for the other modes of operation. With these calibrations capacity data can be entered on the log sheet.

A feature of the testing method is the use of hourly summaries. Due to the *on-off* operation of the heat pump it is necessary, for each hour, to determine separately the kilowatthours for each mode, and also the corresponding Btu produced. Average temperatures for the hour are readily determined. These hourly summaries are then recorded as punched holes in business machine cards,

as shown in Fig. 14. On this card the following data are included: installation number; hour; total kilowatthours consumed by heat pump; kilowatthours consumed by defrosting heater; kilowatthours on each operating mode; average outdoor dry bulb; average outdoor dew point; average indoor dry bulb; average indoor dew point; water temperature in (if used); water temperature out (if used); Btu delivered at each mode; complete history of cyclic operation for the hour.

The final step is to utilize these cards to obtain performance correlations between various factors. For instance, with a group of these cards covering a long period of operation, the following can be calculated by automatic business machines: ***a.*** CP for each operating step *vs.* outdoor conditions; ***b.*** performance factor; *c.* kilowatthours *vs.* degree days; *d.* load factor *vs.* time; *e.* relation between recorded outdoor conditions and records of nearest weather station; *f.* frequency of operation of various steps at different conditions; *g.* frequency of defrosting and required power consumption at all climate conditions; *h.* correlation of factors affecting building heat loss; *i.* relation between electrical demand and outdoor temperature and humidity; *j.* heating/cooling ratio *vs.* various climate parameters; and *k.* correspondence with analytical methods of predicting performance.

CONCLUSIONS

This paper has presented methods for evaluating heat pump performance, both analytically and experimentally. Its nature does not permit the usual quantitative conclusions. However, the following indicate general results:

1. It *is* possible to predict the performance of a heat pump system. This is true even in an air-to-air type heat pump which seeks a new set of operating conditions for each variation of external conditions. Although a complete analysis of the latter is somewhat complex, this kind of treatment is certainly justified in the design of a packaged heat pump which is to be manufactured in quantity and applied under widely differing conditions.

2. Accurately-taken test data are needed to supplement performance analysis. It is particularly important in field testing to obtain sufficient data to provide an accurate indication of *actual* performance—rather than to take only cursory measurements that do not yield true performance.

3. Contrary to common belief, the use of outdoor air as a source of heat for the heat pump is suitable in other than extremely mild climates. The performance analysis methods that have been presented illustrate a representative climate and show that, if supplementary heat can be made available for the relatively short periods of extremely low temperatures, the air-source heat pump can do an excellent heating job at reasonable economy.

4. The advent of the heat pump requires additional consideration of the factors of weather and building construction and their relation to heating capacity required. It appears evident that new methods must be formulated for determining required heating capacity. Such factors as heat lag and thermal storage in buildings need consideration. These methods may well be based upon the system of analysis presented here.

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DISCUSSION

P. R. ACHENBACH, Washington, D. C. (WRITTEN): The paper presents a method of heat pump analysis and a method of gathering field performance data that is interesting and useful.

It is noted that the authors reversed the usual process of selecting heating and cooling equipment of the proper capacity for a house. The conventional method is to compute the heating or cooling load of a structure and then select equipment having equal or greater capacity. For the method described in this paper the authors determine the capacity of a certain model of heat pump and then select the size and type of houses and the climatic zones that would be suited to it. This approach is probably desirable from the heat pump manufacturer's standpoint, but may not be as useful to a building contractor.

A more detailed discussion of the saving in first cost effected by using city water as an auxiliary heat source for the lowest outdoor temperature range would be of interest. For the example cited, how much could the first cost of the heat pump equipment be reduced by using $10.00 worth of water as an auxiliary source of heat? What percentage of the total degree days at the lower end of the outdoor temperature range should ordinarily be supplied by the auxiliary heat source? Or is the magnitude of the auxiliary heat source related to the increment of compressor capacity between two adjacent models in a line of heat pumps? It would appear that, if an auxiliary heat source is economical in one size house, it would also be economical in another size house in the same climate except for the obvious difficulty of finding a heat pump that would have exactly the right capacity for every size house.

The design procedure in this paper does not mention the extra capacity needed for picking up the temperature in a house in the morning after night setback. Whenever an appreciable amount of electricity is used for any method of heating a house, it is desirable on the one hand to cut down the consumption of electricity by letting the house cool off at night, but on the other hand this practice may result in a higher demand charge because of the sustained high demand of electric power during the picking up period. Thus the consumer has two choices, either of which may result in higher electrical costs. It should also be pointed out that although the thermal lag of a house reduces the maximum heating capacity required, the coldest hour at night frequently occurs at daylight or a little earlier so that the maximum heat loss of the house may often take place at about the same time as the house temperature would ordinarily be picked up in the morning. Furthermore, the extra quantity of heat required to raise the house temperature in the morning would be required when the heat pump was operating at its lowest coefficient of performance.

The paper indicates that the field data are taken photographically immediately after the compressor starts or stops. It is doubted that the indicated suction and discharge pressures recorded at these instants would be a good basis for computing the average performance of the heat pump during the entire running period. Has the relation between the photographed pressures at the instants of starting and stopping and the average pressures during the cycle been studied?

C. S. LEOPOLD, Philadelphia, Pa.: I would like to ask the authors if they recorded the internal energy from the illumination and appliances, and whether they took account of the internal heat gains in all of these tests?

AUTHORS' CLOSURE: Mr. Leopold inquires if we are recording internal energy from illumination and appliances, and if we are taking these gains into account. Our testing method does not give us this gain directly, but permits us at least to evaluate it indirectly. Records are being kept of the additional electrical usage in each structure so as to obtain some measure of this gain. Also, since the testing method provides for a fairly precise indication of the heat produced by the heat pump during any set of conditions, we are able to detect variations in the building heat requirements caused by internal heat gains. These variations have been found to be small due to the fact that none of these particular installations has a very substantial internal gain such as is encountered on large commercial installations with high lighting, cooking and occupancy sources.

Mr. Achenbach pointed out that perhaps we were reversing the usual procedure here by applying a given heat pump system to a specific climate, and then determining what size structure it can handle. This, however, is exactly the problem that faces any manufacturer in designing a finite number of sizes of packaged units for application in different types and sizes of structures in widely varying climates. In fact, even for the case where a system is to be custom built for one specific application, the sizing and performance of the system cannot be finally established until a combination of tentatively-sized components is subjected to such an analysis to determine how they will perform. Of course, another method is to size the components on the basis of a few *design* conditions, and then wait until the unit is built and has been operated for a considerable period of time to determine its exact performance under the full range of conditions encountered.

Mr. Achenbach has suggested a more detailed treatment on the aspect of an auxiliary heat source for an air-to-air heat pump. Reference to the top curve of Fig. 6 in the paper reveals that, for the conditions illustrated, the capacity of a straight air-to-air heat pump would have to be approximately doubled to supply sufficient heat at 0 F if an auxiliary source is not used. It is hard to see how this can be justified for the 30 hr a year below 15 F (Fig. 7) during which the increased size is required to supply the 1.9 million Btu (Table 2), or $\frac{1.9}{97.9} = 1.94$ percent of the total degree-days. A half-size heat pump can still provide sufficient heat for 15 of the 30 hr, and the ten dollar expenditure for auxiliary water supply will enable it to supply the remainder. Of course, detailed economic considerations for each individual case, taking into account incremental heat pump sizes available, may vary these figures for the same climate; but, in general, an auxiliary heat source will undoubtedly be the most economic solution except for climates with design heating temperatures of 15 F or above.

The problem of night setback *vs.* no night setback for heat pumps has not yet been settled. However, data are being taken in the testing program described in the paper under both situations in order to evaluate this factor.

The relation between the photographed pressures at the instant of stopping a cycle and the average pressure during the cycle has been studied and they have been found to agree very closely. The operating cycles are usually long enough for the refrigerant pressures to stabilize within a comparatively short time, and any error in assuming them to be constant throughout one running cycle is in the direction of accounting for slightly less capacity than is actually produced.

1383

No. 1383

CONDENSATION ON PREFABRICATED WALLS

By E. R. QUEER* AND E. R. MCLAUGHLIN**, STATE COLLEGE, PA.

THE WALLS of a dwelling are designed to be a barrier between the elements of the weather and the occupants. By trial and error and a little objective thinking satisfactory combinations of materials were discovered. Certain combinations were acceptable for many years until changes in living habits and economic values necessitated alterations in the wall structures. Out of consideration for comfort, the use of insulating material within the wall seemed desirable. The reduction in the cost of fuel for heating was a type of amortization for the cost of including the insulation. Comfort and fuel costs also required that the structure be as tight as possible to reduce infiltration losses and drafts. In many cases the obvious advantages of such procedures overshadowed the lurking danger of condensation.

Consequently, many structures were completed without any thought of providing adequate protection for the structure against the moisture released within the house. When the problem became severe enough to attract engineering attention, the cause was recognized and remedial measures were suggested. As new materials become available and as new uses for common materials are proposed, they must be examined for possible difficulty with condensation. Since corrective measures can be costly, every effort should be directed toward avoiding all objectionable condensation when utilizing new materials or new combinations of materials. It is mandatory, therefore, that the mechanism of water vapor transfer be recognized by those interested in any responsible phase of the building trade.

There are so many facets to the condensation problem that field observations are not sufficient in appraising the performance of a typical construction even though field performance is the ultimate criterion. Surveys of conditions imposed on house structures have been very helpful in formulating the conditions to be

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Presented at the 56th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Dallas, Tex., January 1950.

used in laboratory tests. Additional survey work may be necessary to correlate laboratory results with field results.

Laboratory tests were designed to reduce to a common denominator the conditions imposed upon the test walls. Insofar as possible, all walls of a series, approximately a dozen, were simultaneously subjected to test conditions of moderately severe magnitude. Time of exposure to prescribed conditions was deemed to be equal to or more severe than conditions likely to be encountered in service.

TABLE 1—SUMMARY OF TEST CONDITIONS

	Inside		Outside		Time Days	Vapor Pressure Difference Lb/Sq In.
	Temperature F	RH %	Temperature F	RH %		
A	70	46	20	78	20	0.125
B	70	38	0	73	16	0.125
C	70	38	0	73	29	0.125
D	80	38	0	73	11	0.178
E	70	52	20	78	14	0.150
F	70	52	0	73	26	0.150
G	70	45	30	70	14	0.100
H	70	40	20	60	14	0.115
I	70	40	0	60	14	0.130

The Thermal Research Laboratory of the Engineering Experiment Station at The Pennsylvania State College has a Climatometer with facilities for subjecting a number of full size test panels to a range of temperatures and humidities.

TEST APPARATUS

Panels were set around a platform, 14 × 17 ft, in an insulated test room 20 × 30 ft with a height of 17 ft. All panels were full sized samples supplied by the manufacturer. Temperature outside the test house and temperature and humidity inside the test house were under automatic control. Temperatures were measured at 12 locations on and within the wall. Permanently installed brass points were used for electrically determining the moisture content of sheathing and siding. Dew points of air mixtures were determined by dew-point cup.

Visual observations for condensation in the cavity were made through access doors in the exterior covering of the wall. Doors at top and bottom gave some hint as to where the condensation was forming, if any. In certain cases, alterations were made as deemed advisable after initial tests were complete. See Table 1 for a summary of test conditions.

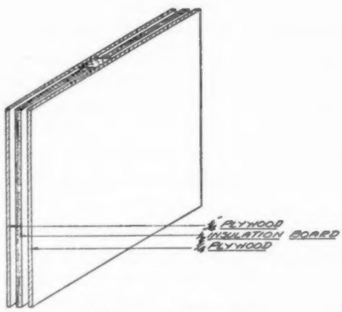
Wall Panel 1 was constructed with ½-in. insulation board between 2-in. framing members. The interior and exterior finish was ¼-in. exterior grade fir plywood. The interior surface was painted one coat resin sealer and two coats inside white paint. The exterior was covered with two coats outside white paint. This wall showed no evidence of condensation under conditions A, B, C and D. This was attributed to the effective interior sealer and paint coat and

also in a small measure to the ventilation of the cold air space to the weather through holes $\frac{1}{4} \times 5/32$ in. provided for the purpose. This is relatively little ventilation, averaging 1/50 sq in. per linear foot of wall.

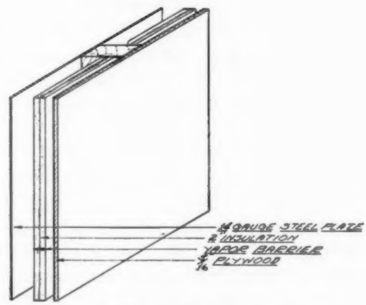
Wall Panel 1a was a variation of Wall Panel 1 in that the interior finish was removed with paint remover. Condensation was not observed under conditions C and D.

Wall Panel 2 with a sheet steel exterior seemed to violate good engineering requirements. Insulation was a nominal 2 in. blanket with vapor barrier. Interior finish was 5/16-in. rough Douglas fir plywood with two coats of lead and oil paint. The sheet steel exterior was an excellent barrier which showed evidence of condensation under condition A and considerably more condensation under condition B.

It was believed that ventilating the cold space where condensation was found would permit what vapor was passing through the paint, plywood, and barrier



WALL PANEL 1



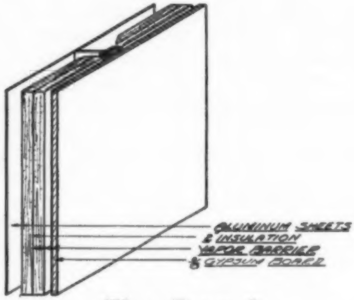
WALL PANEL 2

on the blanket to pass on through to the weather. Accordingly, 1-$\frac{1}{8}$ in. diameter holes, 1 sq in. each, were cut in the steel exterior at the top and bottom to provide 1 sq in. per linear foot of wall, top and bottom. No evidence of condensation was found in the cavity of this wall after exposure to conditions C and D.

At the sill the metal exterior turned in to meet the interior plywood finish. Condensation was noted on the room side of the plywood under all conditions A, B, C and D.

Wall Panel 6 was constructed in such a way that the returned edges of the exterior metal siding served as a stud to hold a wood nailing strip that supported two inches of insulation and $\frac{3}{8}$-in. gypsum board. One layer of insulation extended down over the inside surface of the metal sill channel to insulate what would otherwise be a cold surface behind the baseboard. The gypsum board was foil backed to provide an excellent vapor barrier. There was a vapor barrier on each 1 in. layer of insulation. The cold cavity was vented through slots $\frac{1}{4}$ in. wide and 2 in. long and spaced to provide 1 sq in. of opening per linear foot of wall top and bottom.

It is to be noted that this wall was well provided with vapor barriers well applied and ample ventilation to the cold cavity. All edges of the insulation were



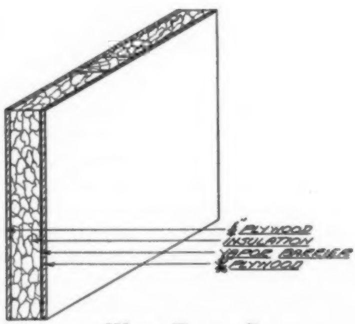
WALL PANEL 6

secured, vertical edges were tacked to the wood strips in the metal studs, and horizontal edges top and bottom were cemented to the metal framing channels.

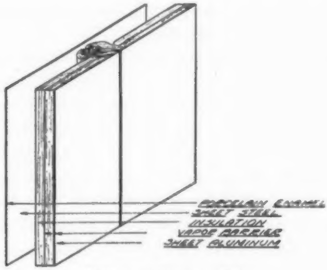
No evidence of condensation was observed under conditions A and B. The ventilation slots in the exterior metal sheet were sealed and the test was continued under conditions C and D. There was no evidence of condensation within the wall cavity. The metal return forming the stud provided a path of low resistance to heat flow and created a relatively cold area on the interior surface directly over the nailing strips. The nails used in fastening the gypsum board to the wood strip were badly rusted.

Wall Panel 7 was of stressed plywood design, using ¼-in. plywood for the interior and exterior finish, with framing members 2-¾ in. thick. Framing members were horizontal on 16 in. centers. Panels were 40 in. wide. One coat of varnish sealer, one coat of undercoat, and two coats of oil paint provided the finish for the exterior and interior. This provided an excellent vapor barrier on both surfaces to supplement the barrier paper on the mineral wool. The insulation filled the 2-¾ in. space between plywood faces.

No evidence of condensation was observed at any point in this wall under conditions A, B, C and D.



WALL PANEL 7



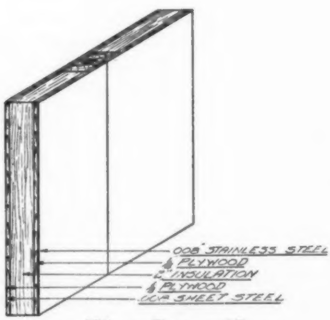
WALL PANEL 14

The paint was removed from a section of the interior face of this panel to expose the ¼-in. plywood. No evidence of condensation was found in this section after exposure to conditions C and D.

Wall Panel 14 was a metal wall in which the aluminum panels forming the interior surface had four inch flanges which clipped together to form the studs. Two inches of insulation were cemented to the cavity side of the interior panels. Porcelain enameled sheets 2 × 4 ft were applied horizontally with all joints sealed with caulking compound. Ventilation to the 3 in. cold cavity was about ½ sq in. per linear foot of wall both top and bottom.

No evidence of condensation was observed in this wall under conditions E and F except as follows. After 25 days were completed under condition F, the outside temperature dropped to −10 F and a light layer of frost was observed on the interior surface of the porcelain enameled sheets. It is quite possible that this moisture was released by the hygroscopic insulating material faster than the ventilation could carry it away. The exterior sheets were only 4 deg above the ambient air temperature and did not provide much margin for rapid changes in ambient temperature.

Wall Panel 17 had thin sheet steel bonded to ¼-in. plywood with thermosetting adhesive. The two inch space between plywood faces was filled with

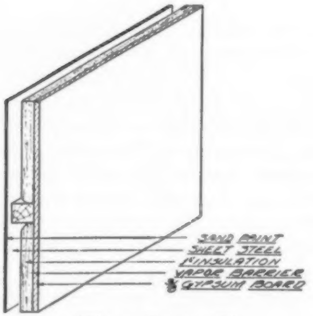


WALL PANEL 17

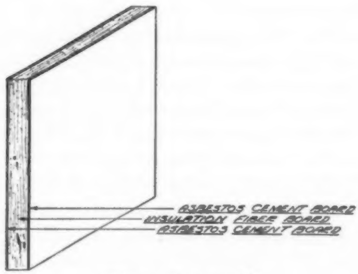
cotton insulation. This formed a wall with an excellent vapor barrier on each surface and hygroscopic material between.

No evidence of condensation was observed under condition E. After exposure to condition F, frost was observed on all access doors. Slight condensation was also observed at the top and bottom of the joint in the interior finish. The thin sheet steel was turned outward at this point to cover the framing member, thereby producing a path of low resistance to heat flow.

Wall Panel 20 had an exterior surface of galvanized sheet steel in tension, primed, covered with exploded slag, and painted. Studs were metal, 39 in. on centers. Horizontal wood furring strips held the 1 in. blanket insulation with a vapor barrier on each face. Interior finish was ⅜-in. plain gypsum board. The cold cavity was ventilated by openings of 2-¼ sq in. per linear foot of wall.



WALL PANEL 20



WALL PANEL 21

No evidence of condensation was observed in the cavity of this wall under conditions E and F. Under both conditions, however, condensation was observed on the heads of the nails holding the ⅜-in. gypsum board. Condensation was also observed at the four corners of the panel where a heavy flange on the framing member turned in to approach the gypsum board and cause a cold spot at each corner.

Wall Panel 21 was a sandwich wall and had no cavity. Two sheets of ⅛-in. asbestos cement board were applied with bituminous adhesive to opposite surfaces of 1-5/16 in. insulating board core. All edges were sealed with a coat of special paraffin. No paint was applied to either inside or outside surfaces.

This wall showed no evidence of condensation under conditions E and F until the ambient temperature dropped to —10 F on the 26th day of condition F. At —10 F there was a 1 in. wide band of condensation along the lower edge of the panel. Following tests at conditions E and F the average moisture content of the wall was 12.2 percent. Visual inspection indicated that moisture accumulation was heaviest in the outer ¼ in. of the fiber board core.

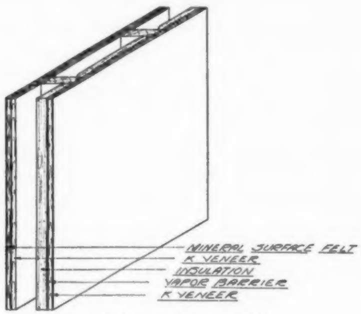
Wall Panel 22 was a sandwich wall similar to Wall Panel 21 except that the outside surface was painted with one coat of cement water paint and the inside was painted with one coat of Vinyl type primer and two coats of alkali resisting semigloss paint. No condensation was observed under condition E. When the

temperature dropped to —10 F on the 26th day of condition F, beads of condensation were evident 6 in. from the floor at the center of the panel and up 4 ft at the edges of the panel. Average moisture content was 5.8 percent at end of test.

Wall Panel 23 was similar to Wall Panel 21 except that the outside was covered with one coat of cement water paint and the inside was painted with one coat of base coat oil paint and two coats of semigloss paint.

No condensation was observed in this panel under conditions E and F until the temperature dropped to —10 F on the 26th day of condition F. At that time condensation appeared on the bottom 6 in. of the panel and up 2 ft at the joint. Average moisture content was 3.5 percent.

Wall Panel 24 was the same as Wall Panel 22 except that the sample contained a "T" type wood stud joint with a wood batten. Condensation was noted at the foot of the joint under conditions E and F.



WALL PANEL 26

Wall Panel 25 was similar to Wall Panel 22 except that the joint was an assembly of two pieces of ¼ in. × 4 in. × 8 ft asbestos cement board acting as battens and attached by ¼-in. bolts through the wall sample on 12 in. centers.

Heavy condensation was observed on the nuts and bolts at the joint under conditions E and F.

Wall Panel 26 was fabricated with 1 × 3 in. studs on 8-½ in. centers covered with 3/16 in. Douglas fir veneer. The interior veneer was treated on the cavity side with a trowel coat of high melting point asphalt, including one inch return on the face of the studs. One inch of insulation was stretched from top to bottom and held in place by cleats and the asphalt. This provided a 1-¼ in. cavity on the cold side of the insulation. The veneer surfaces of the panel were covered with a bonding coat of urea adhesive and 30-lb kraft paper. The exterior surface was protected by a bonding coat of casein glue and 90 lb (per 100 sq ft) mineral surfaces roofing. No paint or wall paper was applied over the interior veneer and kraft paper.

No evidence of condensation was observed in this panel under conditions E and F.

Wall Panel 27 was similar to Wall Panel 26 except that a sheet of felt weighing 15 lb per 100 sq ft and highly saturated with asphalt was added between the 1 in. insulation blanket and the trowelled coat of high melting point asphalt.

No evidence of condensation was observed in the cavity of this wall after exposure to conditions E and F. After 19 days exposure to condition F, the kraft paper on the interior surface showed damp spots just above the floorline. After 26 days exposure to condition F, the same area was damp and spotted with a blue green mold, apparently feeding on organic material used in the fabrication of this wall.

Wall Panel 28 was similar to Wall Panel 26 except that a sheet of felt impregnated and coated with asphalt and weighing about 17 lb per 100 sq ft was added between the 1 in. insulation blanket and the trowelled coat of asphalt.

The performance of this wall was identical to that of Wall Panel 27 except that the area of dampness was greater (up to 5 in. above the floor) and the amount of mold was greater.

Wall Panel 29 was similar to Wall Panel 26 except that, in lieu of the trowelled-on coat of high melting point asphalt, the vapor barrier was a sheet of paper impregnated and coated with asphalt and weighing 11 lb per 100 sq ft with edges glued and tacked to studs.

This wall performed in much the same way as Wall Panel 27. The interior surface of the kraft paper was damp 3 in. above the floor although the mold was not as severe as it was on Wall Panels 27 and 28.

Wall Panel 30 was similar to Wall Panel 26 except that, in lieu of the trowelled-on coat of high melting point asphalt, the vapor barrier was a sheet of asphalt saturated felt weighing 15 lb per 100 sq ft, with edges glued and tacked to studs.

Observations on this wall were similar to those made on Wall Panel 26. The interior surface was wet 3 in. above the floor line with a few spots of mold on the damp area.

Wall Panel 41 was constructed of metal frame with 1 in. mineral wool board insulation between studs. The exterior finish was slater's felt and painted aluminum clapboards. The interior finish was metal lath and plaster with two coats of water paint.

Condensation formed very quickly on the metal siding under condition G. It was observed that the panel had been plastered five months prior to the tests, but the plaster was not considered dry at the beginning of the tests. Water paint is not considered to be a vapor barrier and in the absence of any other barrier considerable moisture might flow through the plaster and insulation to the cold siding. It is believed that moisture was removed from the plaster, too.

After this wall was exposed for one week to condition H, heavy frost was observed on the inner surface of the lower siding. At that time vent holes were drilled through the siding and the sheathing paper. Six 9/16 in. diameter holes were drilled 6 in. on centers to provide ½ sq in. vent area per linear foot of wall. An additional week's exposure to condition H produced no net change in the quantity of condensation.

After this wall was exposed two weeks to condition I, the quantity of condensation on the siding was reduced so that frost covered 50 percent of the area of the observation ports.

Condensation appeared on the warm surface of this wall up to 15 in. above the floor level. It was observed that there were considerable plaster droppings at the foot of this wall.

Wall Panel 42 was similar to Wall Panel 41 with the substitution of two coats of aluminum paint for the two coats of water paint on the interior surface. This wall was plastered five months prior to the beginning of tests. This panel showed very little condensation under conditions G and H, and no vent holes were drilled in the siding. After this wall was exposed to condition I, frost covered 50 percent of the area of the observation doors. Condensation occurred on the room surface up to 18 in. above the floor level.

Wall Panel 43 was similar to Wall Panel 41 with the exception that the 1 in. insulation had been omitted. The performance of this wall was similar to that of Wall Panel 41 with the exception that condensation was more marked on the lower part of the sheathing and the lower part of the interior finish. Condensation extended upward 20 in. on the room surface. Ice was evident opposite one stud and the bottom channel.

Wall Panel 44 was similar to Wall Panel 41 except that an aluminum foil covered kraft paper was installed between the insulation and the plaster base. This barrier was not fastened at its edges. Plastering was completed only 10 days prior to the beginning of the tests.

Condensation formed very quickly on the metal siding under condition G. After this wall was exposed for one week to condition H, heavy frost was observed on the lower access door. Vent holes were drilled through the siding to provide ½ sq in. of vent area per linear foot of wall. After the wall was exposed for an additional week to condition H, it was evident that the frost was evaporating. Heavy frost accumulated on the upper access door during two weeks exposure at condition I.

DISCUSSION OF RESULTS

The results of laboratory tests are enhanced when they can be interpreted for practical use. The tests reported herein were intended to determine the performance of a construction when subjected to moderately severe conditions. Time was a factor and a reasonable acceleration was desired. Acceleration could not be carried too far, however, so that a 14-day test period seemed to be the minimum time of test. The sequence of exposure for any test panel was from moderate to severe so that the performance under moderate conditions could be observed first. In this way, a panel which failed under moderate conditions could be rejected or revised for tests under more severe conditions. On the other hand, those panels which withstood testing under moderate conditions were seasoned to a certain extent before beginning the more severe tests.

The purpose of the testing program was more than a comparison of a number of constructions. The purpose was to show a measure of performance for each panel under service conditions. To help evaluate the results of the test series, a number of criteria were suggested and developed during the test program. It was deemed necessary to keep the criteria applicable to wall constructions which might be used in small tightly constructed dwellings. For this reason, the failure of a type of construction to meet any one criterion does not eliminate the construction from consideration for larger buildings with warm floors and relative humidities lower than the humidities used in the tests. Likewise, a summarized

TABLE 2—PERFORMANCE OF SOME OF WALLS TESTED

Wall Panel	Test Condition	Condensation in Cavity Criterion 1	Condensation on Warm Surface Criterion 2	Cavity Dew-Point Criterion 3	Average Dew-Point Criterion 4
1	A	none	none	—	—
	B	none	none	OK	OK
	C	none	none	OK	OK
	D	none	none	OK	OK
1a	C	none	none	OK	OK
	D	none	none	OK	OK
2	A	condensation	—	—	—
	B	condensation	condensation	fail	fail
2a	C	none	condensation	OK	OK
	D	none	condensation	OK	OK
6	A	none	none	—	—
	B	none	none	—	—
6a	C	none	none	—	—
	D	none	none	—	—
7	A	none	none	—	—
	B	none	none	OK	fail
	C	none	none	—	—
	D	none	none	—	—
7a	C	none	none	—	—
	D	none	none	—	—
14	E	none	none	—	—
	F	none	none	—	—
17	E	none	none	—	—
	F	condensation	condensation	—	fail
20	E	none	condensation	—	—
	F	none	condensation	—	—
21	E	none	none	—	—
	F	none	none	—	—
22	E	none	none	—	—
	F	none	none	—	—
23	E	none	none	—	—
	F	none	none	—	—
24	E	none	condensation	—	—
	F	none	condensation	—	—
25	E	none	condensation	—	—
	F	none	condensation	—	—
26	E	none	none	OK	OK
	F	none	none	OK	OK

TABLE 2—PERFORMANCE OF SOME OF WALLS TESTED *(Continued)*

Wall Panel	Test Condition	Condensation in Cavity Criterion 1	Condensation on Warm Surface Criterion 2	Cavity Dew-Point Criterion 3	Average Dew-Point Criterion 4
27	E	none	none	OK	OK
	F	none	condensation	OK	OK
28	E	none	none	OK	fail
	F	none	condensation	OK	OK
29	E	none	none	—	—
	F	none	condensation	—	—
30	E	none	none	—	—
	F	none	condensation	—	—
41	G	condensation	none	fail	fail
	H	condensation	none	fail	fail
41a	H	condensation	none	fail	fail
	I	condensation	condensation	fail	fail
42	G	condensation	none	fail	fail
	H	none	condensation	fail	fail
	I	condensation	condensation	fail	fail
43	G	condensation	none	fail	fail
	H	condensation	none	fail	fail
43a	H	none	none	fail	fail
	I	condensation	condensation	fail	fail
44	G	condensation	none	—	—
	H	condensation	none	—	—
44a	H	condensation	none	—	—
	I	condensation	condensation	fail	fail

statement of the performance of a wall cannot include a qualified statement of the degree of divergence from an arbitrary criterion. For this reason, slight traces of condensation that were judged to be unimportant were ignored and the summary indicates that there was no condensation. The term condensation cannot indicate the degree of accumulation of frost or water and was therefore used for all cases where the quantity of condensation was judged to be significant to the performance of the construction. Fortunately, there were not many borderline cases where the degree of condensation was such that a positive rating was in doubt.

JUDGING WALL PERFORMANCE

The performances of some of the walls tested are summarized in Table 2. The following criteria were used in judging the performance of the walls under the conditions of the tests.

Criterion 1. There shall be no evidence of condensation within the cavity of the wall, or between any of the various layers of materials making up the sample after 14 days' test.

Criterion 2. There shall be no evidence of condensation on the room surface of the wall after 14 days' test.

Criterion 3. A study of reliable psychrometric data shall indicate that the dew-point of the air in the cavity shall always be lower than the coldest surface temperature in contact with it.

Criterion 4. A study of reliable psychrometric data shall indicate that the dew-point of the air in the cavity shall average at least 5 deg below the coldest surface with which it comes in contact during the 14-day test period.

CONCLUSIONS

This series of tests indicates that several conditions should be avoided in prefabricated construction, particularly in construction using materials having low resistance to heat transfer, such as metal or concrete. Panels frequently met all requirements for satisfactory walls except that condensation occurred on the interior surface near a metal stud, metal sill, or a returned edge of an exterior metal sheet. In other cases, excessive plaster droppings within the wall created a path for high heat transfer and contributed to the formation of condensation on the interior plaster surface.

The advantages of a metal exterior are frequently offset by the undesirable accumulation of condensation. As has been demonstrated on Wall Panel 2, this feature may be overcome in a measure by adequate ventilation from the weather to the wall cavity.

The use of a good vapor barrier well applied on the warm side of the wall is emphasized. Trouble with condensation in the wall cavity was seldom experienced where an effective vapor barrier was properly used. Plywood served as a good vapor barrier, especially if it was painted with several coats of oil paint. The walls having metal panels as the interior finish were very resistant to vapor flow.

The joints between fabricated panels frequently represent an area of condensation. Where bolts or metal strips are used through metal, contact points present a cold surface to the humid air in the heated space. Several panels showed condensation at joints only, whereas the remainder of the panel was suitable for the conditions encountered.

New combinations of materials must be examined to assure that substances which may develop fungus or corrosion in the presence of humidity and warmth are properly protected by collateral materials or techniques to avoid deterioration.

ACKNOWLEDGMENT

The series of tests described herein were sponsored by the Housing and Home Finance Agency, Washington, D. C. Prefabricated panels were obtained by the Agency in cooperation with the manufacturers of the respective panels. Test conditions and criteria were suggested by R. R. Britton, structural engineer, Housing and Home Finance Agency.

DISCUSSION

SABERT OGLESBY, JR.,* Birmingham, Ala.: Can the authors tell me whether the instrument field was surveyed to determine if there were any instruments which would evaluate the extent of condensation?

ART THEOBALD, El Paso, Tex.: Three questions occur to me: (1) Is not the condensation on the windows an early or an immediate indication of condensation in the walls? (2) Is any thought given to the real necessity of adding humidity through heating appliances? (3) Should not some thought be given in actual applications to bringing in outside air of a lower moisture content? This might be accomplished by means of the heating system or kitchen ventilation.

W. A. DANIELSON, Memphis, Tenn.: Could the authors be more specific on the climates that were mentioned? The climate in Florida is different than on the west coast, which ranges from the arid climate of San Diego to the very humid climate of the state of Washington.

WARREN VIESSMAN, Port Hueneme, Calif.: This paper is very interesting. It has a lot of applications not only in the particular type of residences to which we might be accustomed, but in trailers, airplane construction, and in prefabricated huts for Arctic territories, military purposes, and things of that sort. I wonder if Professor Queer might say something as to the provision for the vapor barrier in such constructions; and if he might add a few words in regard to honeycombed constructions and metal surfaces.

F. E. INCE, St. Louis, Mo.: I am wondering, in lieu of ventilation or washed air space, if perhaps a low vapor panel or pressure could be used for collection of moisture.

AUTHORS' CLOSURE: There are numerous methods which can be used to detect condensation but no one of them will evaluate the extent of condensation. The cost of applying a large number of small detectors over the wall surface is prohibitive and, furthermore, the introduction of a lot of gadgets alters the temperature and air flow patterns within the wall. A photograph is helpful but very difficult to obtain since the strong lighting alters the temperature conditions and quite frequently the wall must be partially disassembled, again altering the temperature conditions. The absence or presence of condensation is best observed by eye and the extent must be evaluated by a quick estimate of the area affected.

The appearance of condensation on windows is not a reliable indication of condensation in the walls. It is quite possible that the walls are so constructed that they can endure a much higher humidity condition than the windows. In that case the windows act as condensers and prevent the humidity from rising to the critical condition for the walls. On the other hand the walls may be poorly constructed so that the humidity can migrate through the wall to condense on a cold surface. In this case the windows may or may not show condensation and are practically useless as indicators of what is happening within the walls. Time of exposure is also a factor. Severe humidity of short duration may condense extensively on a window but thermal and moisture lag would prevent condensation within the wall.

There is considerable doubt of the value in adding humidity through heating appliances or humidifiers unless the structure was designed and is capable of withstanding an artificially maintained humidity.

The use of ventilation to reduce humidity is used frequently and can be quite helpful in alleviating a serious condensation problem.

Climate is a rather important factor in the condensation pattern. Temperature and prevailing humidity are two prime elements. Any attempt to correlate climate and

* Southern Research Institute.

geography is of necessity a compromise. This is evident in the circuitous boundaries between zones of different design temperatures. The realization that local climate has a bearing on such things as heating and condensation gave impetus to the science of micro-climatology.

The challenge to provide an adequate vapor barrier in prefabricated construction is very real. Among the other requirements of strength, weight, and portability the need for a good vapor barrier is likely to be overlooked or neglected, especially where panels are joined. Spacing members having a low resistance to heat flow may contribute to the condensation problem. Homogeneous cores or honeycombed constructions are attempts to avoid these difficulties. A metal surface will prevent the migration of water vapor into the wall, providing the seams are satisfactory. With sufficient insulation the interior wall surface will remain warm enough to prevent condensation.

There are numerous ways of controlling humidity and vapor pressure but in the case of a wall in a house during the winter time ventilation appears to be very simple and economic. That is assuming, of course, that a vapor barrier is not adequate or practical.

No. 1384

SOLAR HEATING OF HOUSES BY VERTICAL SOUTH WALL STORAGE PANELS†

By ALBERT G. H. DIETZ* AND EDMUND L. CZAPEK**, CAMBRIDGE, MASS.

IN the original M.I.T. solar house constructed in 1937, solar energy was collected in flat-plate collectors, built into the roof, and was stored in a large tank of water, situated in the basement, from which it was drawn as needed. This system collected and stored heat during the entire year and required no auxiliary mechanical heat. The installation was designed not to be competitive in cost with ordinary heating systems, but to obtain information respecting the performance of solar energy collecting and storing systems. This work has been reported by Hottel and Woertz[1]. In the present test house the goal was to make the system as simple and economical as possible by combining the collecting, storing, and heating functions in one simple container, and controlling the flow of heat energy by glass plates and curtains or blowers.

In a flat-plate heat-collector designed to operate during the entire year, the angle of the collector with the vertical must be large because the summer sun is high in the sky, but if a collector is expected to be operative only during the heating season, it can be vertical and be nearly as effective as if it were inclined at the most favorable angle for the winter sun. At an angle of incidence of 40 deg, clear glass transmits approximately 95 percent as much solar energy as it does at normal incidence[1]. Consequently, vertical flat-plate collectors can fairly simply be built into the south wall of a building. If the collecting surface forms the face of the heat storage unit no piping, ducts, or other means are required to transfer the heat from the collecting surface to the storage medium. Furthermore, if heat is abstracted from the other side of the heat storage unit, the unit becomes a combined collector, storage medium, and house heater in the simplest possible arrangement of storage wall.

Various materials can be used for the heat-storing medium as discussed in a paper by Telkes[2]. These may be classified as sensible-heat storers and latent-heat storers. Of the former, water is the most efficient of the common materials like concrete, crushed stone, iron ore, or masonry, on a volume basis. Of the latter, Glauber's salt or sodium sulfate decahydrate ($Na_2SO_4 \cdot 10H_2O$) is theoretically promising and has

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[1] Exponent numerals refer to Bibliography.

† Also issued as Publication Number 27 of the Godfrey L. Cabot Solar Energy Conversion Project at Massachusetts Institute of Technology.

Presented at the 56th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Dallas, Tex., January 1950.

often been suggested because of its favorable melting point (approximately 90 F), its high heat of fusion, and consequently high theoretical efficiency as compared with water, and its low cost and availability. In a previous partial heating season (1946-1947) in the test house here described, Glauber's salt gave considerable difficulty because of its tendency to stratify upon cyclic melting and freezing, to overheat, and to undercool, all of which reduced its theoretical efficiency considerably. This behavior corroborated previous small scale laboratory experience. It was decided, therefore,

FIG. 1. EXTERIOR OF TEST HOUSE SHOWING SOUTH WALL

Test cell at left has no storage units. Adjacent six cells have various arrangements of storage units. End three windows are in office space. Aluminum-coated curtains drawn in all cells except cell at left. Horizontal pyrheliometer is mounted on roof and measurements of diffuse radiation are being made.

to return the study of Glauber's salt and other fusible salts to the laboratory and to concentrate upon water during the heating season covered by this paper.

EXPERIMENTAL PROCEDURE

The test house or laboratory has been previously described[3]. Fig. 1 shows the exterior, and Figs. 2 and 3 are the plan and cross section of the house. The solar energy collecting walls (Fig. 2) were oriented due south. The building contained seven test cells, an instrument room, and a guard room (Figs. 2 and 3).

All doors were heavy refrigerator-type doors. The instrument room and the guard room were kept at 70 F and therefore helped to prevent appreciable heat losses through the side walls of test cells 1 and 7. Because the construction was well insulated and because the number of air changes in the test cells was small, the north walls of test cells 2 and 3 were provided with blackened copper panels or heat leaks through which enough heat would be lost to raise

the overall coefficient of heat loss of these two test cells to a figure similar to that of the usual well-insulated house under normal conditions of use. In each test cell the area of the south glass walls was 32.2 sq ft; the combined area of floors, north wall, and ceiling was 136.9 sq ft, and in test cells 2 and 3 the areas of the copper heat leaks were 6.5 sq ft. Small variations in area, not over one percent, existed from cell to cell.

The solar energy or storage wall units shown in Fig. 4 comprised the south wall and combined the functions of collection, storage, and heating the interiors of the test cells. In test cells 1-6 the south walls consisted of a commercial double glass; edge-sealed, 7/32-in. thick, ½-in. air space, figured glass. In test

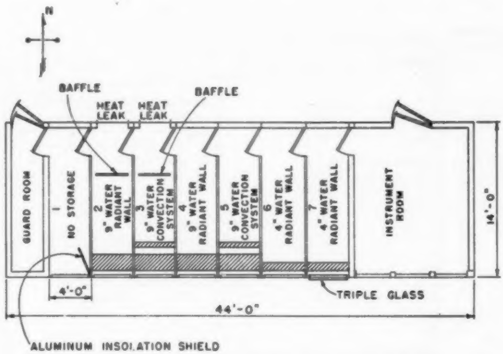

FIG. 2. PLAN OF TEST HOUSE SHOWING ARRANGEMENT OF TEST CELLS

cell 7 the south wall was triple glass of similar manufacture. Figured glass was chosen because it is considerably less costly than plain or plate glass.

Behind the glass walls were water-filled containers consisting of standard 1 gal and 5 gal cans. The 1 gal cans provided a 4-in. thickness and 9-in. height. The 5 gal cans provided a 9-in thickness and 9-in height. Cans were laid up to fill completely the space behind the glass, and were supported at 18-in. intervals in height by wooden shelves to prevent excessive loads on the lowermost cans. The joints between cans, and the vertical joints between cans and window frames were sealed with tape to prevent air leakage. Outer faces of cans were painted a matte black to absorb the highest possible percentage of incident solar radiation. Inner faces were painted white. (Any inner surface color could have been chosen because at the long wave length of radiation from the inner faces almost any surface would act like a black body.)

At night, double aluminum-painted roll curtains with a ¾-in. air space between the layers were drawn between the glass and the containers to reduce outward heat losses. Curtains were up during the day for the eight hours of usable sunlight, and drawn the rest of the time.

The end test cell, number 1, had no solar energy collection units, but it did have a double aluminum-painted curtain similar to the other test cells. It was used as a basis of comparison with the other six test cells and also with the

usual *solar* house which allows the sun's energy to pour through south-facing glass walls in the daytime, but makes no attempt to store that energy except by absorption in floors, walls, and furnishings.

When needed to maintain the temperature of the test cells at the desired level (70 F), heat was transferred from the inner surfaces of the containers either by natural convection and direct radiation from those surfaces or by forced

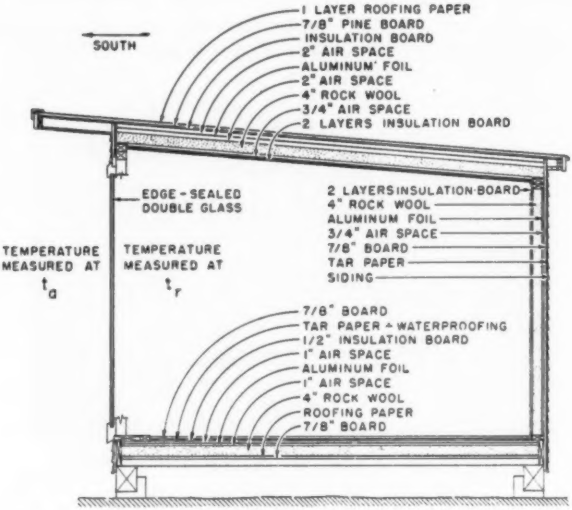

FIG. 3. CROSS SECTION OF TEST HOUSE SHOWING CONSTRUCTION

convection, as shown in Figs. 4A and 4B. Radiation was controlled, as shown in Fig. 4A, by raising and lowering the inner curtain. Forced convection was obtained by a 6-in. blower, (Fig. 4B) which drew room air through an opening near the bottom of an insulated wall interposed between the containers and the test cell, blew the air upward along a baffle, past the inner surfaces of the containers, down the other side of the baffle, and out into the test cell at the floor level. Radiation systems were used in test cells 2, 4, 6, and 7; forced convection systems in test cells 3 and 5.

The seven test cells made possible comparisons among:

1. Storage *vs.* no storage,
2. Various water wall thicknesses,
3. Storage wall performance under different heating requirements,
4. Test cell heating from storage wall as a radiant panel or by a convection system,
5. Triple *vs.* double glass.

Auxiliary electrical heaters of the sunbowl type—574 w average measured

capacity—were placed in the test cells to keep the temperature at the proper level if the solar energy units failed to do so.

A stratification test showed temperature differences as high as 11.7 deg in the quiet air of the test cells. Consequently, 4-in. fans were placed in the test cells to keep the air moving. They were found ample to keep temperature differences in the test cells below 1 deg.

Additional electrical units were: the convection blowers (17.7 watts) in test cells 3 and 5; the motors for raising and lowering the inner shades in test

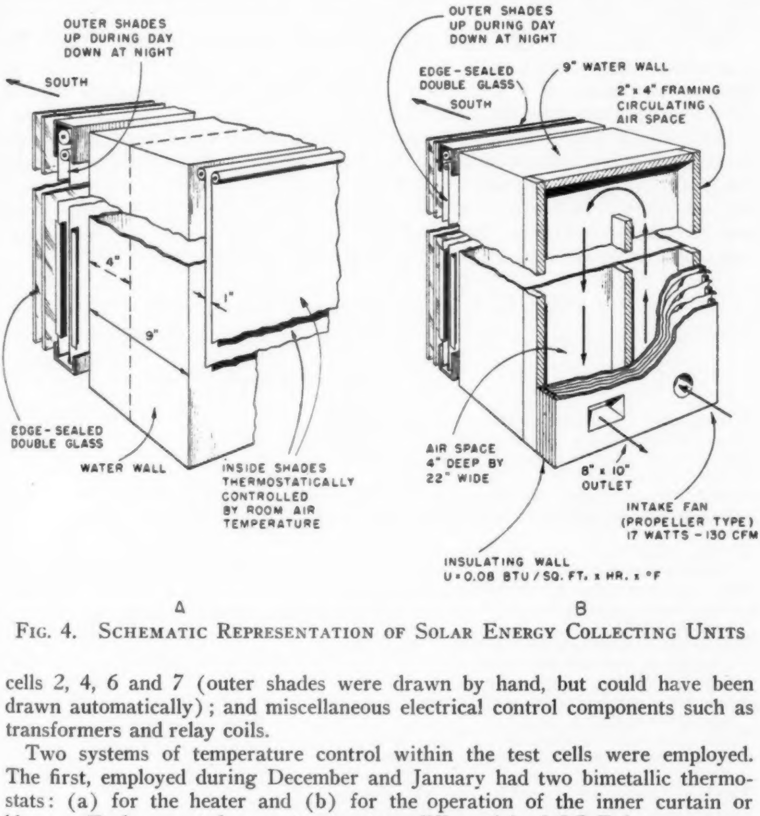

FIG. 4. SCHEMATIC REPRESENTATION OF SOLAR ENERGY COLLECTING UNITS

cells 2, 4, 6 and 7 (outer shades were drawn by hand, but could have been drawn automatically); and miscellaneous electrical control components such as transformers and relay coils.

Two systems of temperature control within the test cells were employed. The first, employed during December and January had two bimetallic thermostats: (a) for the heater and (b) for the operation of the inner curtain or blower. Each operated on a temperature differential of 0.5 F between open and closed positions. This system was not entirely satisfactory. When the temperature dropped below 65 F because the heat storage units were depleted, the sunbowl heaters often caused the controls to overshoot the 65 F lower temperature level and raise the inner curtain, whereupon the heat would be absorbed by the cold containers, sending the temperature again below 65 F. This brought about an unstable condition in which the heater and curtain operated too fre-

quently, the temperature fluctuated too uncertainly, and some of the auxiliary heat actually was used to heat the containers, an undesirable situation. A second system of controls was therefore put into effect during February, March, and April. It involved controls based both on room temperature and on storage wall temperature.

The upper control temperature was increased to 72 F because the motors of the air-circulating fans gave off enough heat to raise the temperatures of the test cells approximately 2 deg. The lower control temperature was raised to 70 F for the same reason, and because 65 F to 67 F was considered to be uncomfortably cool.

In test cell 1 no controls were placed upon the upper temperature. The room was allowed to overheat, as it might in the usual *solar house* unless shades or drapes were drawn or windows were opened. The lower temperature level of 70 F was maintained by the sunbowl heater.

As shown in Fig. 2, baffles were placed in test cells 2 and 3 to prevent direct radiation between storage walls and inner cold surfaces of the blackened copper heat leaks. An aluminum shield was placed in test cell 1 to protect the side-wall adjacent to test cell 2 from direct sunlight. Because test cell 1 overheated, there was some heat penetration through the partitions to test cell 2. This was calculated, and allowance was made for it in the heat balance on test cell 2.

TABLE 1—AVERAGE THERMAL COEFFICIENTS

	THEORETICAL Btu/(sq ft) (hr) (F deg)	EXPERIMENTAL[a] Btu/(sq ft) (hr) (F deg)
Double glass window alone (at average temperature level of 60 F)	0.61	
Double glass window alone (at average temperature level of 35 F)	0.51	
Double glass window alone when part of a glass and curtain combination (at average temperature level of 35 F)	0.57	
Double aluminum curtain alone (at average temperature level of 55 F)	0.24	0.84
Double glass window in combination with double aluminum curtain (at average temperature level of 50 F)	0.17	0.34
Double glass window in combination with double aluminum, and film resistance of collector surface (at average temperature level of 54 F)	0.155	0.29
Triple glass window alone (at average temperature level of 35 F)	0.345	
Triple glass window when part of a glass and curtain combination (at average temperature of 35 F)	0.36	
Triple glass window in combination with double aluminum curtain (at average temperature of 50 F)	0.145	0.25
Triple glass window in combination with double aluminum curtain and film resistance of collector surface (at average temperature of 54 F)	0.135	0.22
Floor, north wall, ceiling (test cells 4, 5, 6, 7)		0.056
Floor, north wall, ceiling (test cells 2, 3)		0.116

[a] At average overall temperature difference (inside room or collector surface to outside air) of 40 F.

Heat-Loss Coefficients

Heat loss coefficients for the double and triple glass, the double aluminum curtains, and the combined floor, north wall, and ceiling were determined by a combination of test and theory.

The starting point for these determinations was the glass. The heat-loss coefficients for the glass were first theoretically calculated by the method of Hottel and Woertz[1]. Several night-time tests were subsequently run in test cell 1, and the electrical power necessary to maintain a temperature of 70 F inside the test cell was measured at the same time that measurements were made of the interior temperature, and temperatures of the air space between the glass

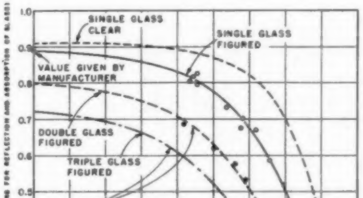

FIG. 5. TRANSMISSION CURVES OF CLEAR AND FIGURED GLASS

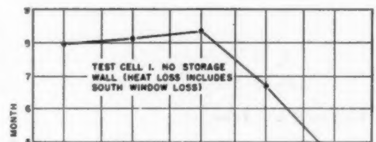

FIG. 6. MONTHLY HEAT LOSS OF TEST CELLS, NOT INCLUDING OUTWARD HEAT LOSS OF COLLECTORS

and the drawn double aluminum curtain, and the outdoor temperature. From the measured temperature drop from the shade-window air space to the outdoors and the theoretically calculated double glass window thermal coefficient, the heat loss during the test runs of the south window (area of 32.2 sq ft) was determined. This south window heat loss subtracted from the total electrical equivalent heat loss of the cell yielded, when divided by the overall temperature drop of inside to outside air, an averaged thermal coefficient for the ceiling, north wall, and floor (area of 136.9 sq ft). It will be referred to as an *experimental* value, in contrast to the *theoretical* value involving no measurements.

The air space between the curtain and window was observed to have a higher temperature value than theoretically supposed. The actual temperature drop through the curtain therefore was measured in conjunction with the window temperature drop. The averaged measured ratio of shade temperature drop to window temperature drop was 0.57 as compared to the theoretical value of 2, and in addition, a varying overall temperature drop (inside to outside air) had a marked influence on this ratio while theoretically, the ratio should have been fairly constant. In the actual case, the close fitting guides at the edge of the

curtain were not effective enough to reduce the intermixing of air around the edges of the curtain to a negligible amount, which was a theoretical condition, although the theoretical window thermal coefficient was adjusted from a calculated value to experimental value as dictated by measured and theoretical temperature ratios.

In test cells 2 and 3 the heat-loss coefficients for the heat leaks were determined from: measured temperature drops from inside air to copper and from copper to outside air; assumed inside surface coefficient based on free convection; and area of copper surface. For heat flow through two parallel heat resistances

$$\frac{t_l - t_o}{t_i - t_l} = \frac{h_i}{h_o} \quad \ldots\ldots\ldots\ldots\ldots \quad (1)$$

where

t_l = temperature of heat leak.
t_o = outdoor air temperature.
t_i = indoor air temperature.
h_i = interior surface thermal coefficient, assumed equal to 2.
h_o = outside surface thermal coefficient.

Utilizing the relation

$$U_1 = \frac{h_i}{1 + \frac{h_i}{h_o}} \quad \ldots\ldots\ldots\ldots\ldots \quad (2)$$

where

U_1 = heat-loss coefficient for the heat leak, the required equation becomes

$$U_1 = \frac{2}{1 + \frac{t_l - t_o}{t_i - t_l}} \quad \ldots\ldots\ldots\ldots\ldots \quad (3)$$

The product U_1A_1 (heat-loss coefficient times the area of the heat leak) was determined and added to the similar product, UA, of the floor, north wall, and ceiling exclusive of the heat-leak area, assuming the value U for cells 2 and 3 to be the same as that determined for cell 1. From the sum U_1A_1 plus UA the overall heat-loss coefficient for the floors, north walls, and ceilings of test cells 2 and 3 was determined.

The heat-loss coefficients so obtained are shown in Table 1.

Measurements

During the period of operation of the laboratory, measurements were made and continuous records kept of the following:

1. *Temperature: a.* Room air temperature of test cells 1-7, inclusive; *b.* Average mid-wall temperature of storage walls in test cells 2-7, inclusive; *c.* Average inside surface temperature of the copper window heat leaks—test cells 2 and 3; *d.* Outside air temperature.

2. *Solar Energy: a.* Horizontal incidence; *b.* Vertical incidence.

3. *Electrical Energy: a.* Electrical input into each room; *b.* Time of heater operation in each room.

Temperatures were recorded with copper-constantan thermocouples. Storage wall temperatures were the average of 10 points at mid-wall. Occasional meas-

urements were also made of the inside and outside surfaces of the storage containers by means of four thermocouples on each surface. These surface temperatures were found to differ by not more than three degrees during periods of maximum solar radiation, and to differ inappreciably during periods of no radiation.

Intensity of solar radiation was measured by two Eppley pyrheliometers mounted horizontally and vertically on the roof of the building. Continuous records were kept on recording potentiometers.

The amount of solar radiation transmitted through the figured glass, was calculated from: (1) direct radiation from the sun; (2) diffuse radiation from

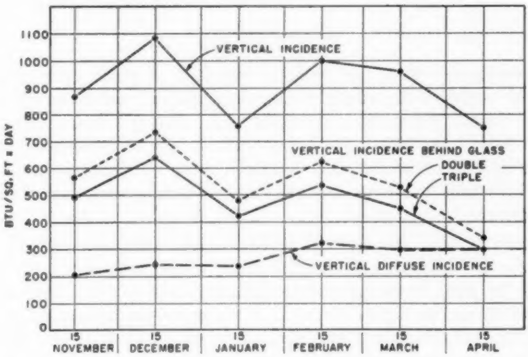

FIG. 7. AVERAGE MONTHLY VALUE OF DAILY SOLAR INCIDENCE (CAMBRIDGE, MASS. 1947-1948)

the sky; (3) experimental determination of the transmissivity of the figured glass at various angles of incidence.

Spot readings of diffuse radiation were made at intervals nearly every day by shading the target of the vertical pyrheliometer for approximately two minutes by a disc of 4.5 in. diameter held 10 disc diameters distant. On days when spot diffuse readings were not available, diffuse values were taken from the solar charts from low points caused by clouds eclipsing the sun temporarily. Diffuse values so found were fairly constant at 30 to 40 Btu per sq ft per hour during the heating season. The higher values occurred during periods of high reflection from freshly fallen snow.

Transmissivity of figured glass was found to average 10.8 percent less than that of clear glass, the actual value depending upon the angle of incidence as shown in Fig. 5 which presents experimental values for single glass, values for double glass calculated from those of the single glass and then verified experimentally, and calculated values for triple glass.

The overall average transmittance for sunlight of a system of two figured glass plates (allowing for both reflection and absorption at various angles of incidence) was: October—0.58; November—0.64; December, January, February—Average 0.64; March—0.53; April—0.47.

TABLE 2—DISTRIBUTION OF DAILY SOLAR INCIDENCE

SOLAR INCIDENCE BTU PER SQ FT PER DAY	OCCURRENCE DAYS	OCCURRENCE, PERCENT OF TOTAL NUMBER OF DAYS
Greater than		
2000	6	3
1500	41	23
1000	83	46
500	118	65

TEST RESULTS

Monthly heat losses from the various test cells are shown in Fig. 6†. Test cell 1 lost heat through the south window in addition to floor, north wall, and ceiling. The others lost heat essentially only through floor, north wall, and ceiling. The high heat losses from test cell 1, whose construction except for storage walls was the same as 4, 5, 6, and 7, were occasioned by the absence of the storage wall and the considerable amount of overheating which occurred during the daytime.

Fig. 7 presents data on vertical incidence of solar energy for the period November—April. This was an exceptionally snowy winter in Cambridge, and the period of heaviest snowfall during January was accompanied by markedly reduced sunlight, as the figure shows. The three-month average vertical incidence for December, January, and February was 945 Btu per sq ft per day, of which 270 were vertical diffuse. The average transmitted and absorbed energy with double and with triple glass was 605 and 530 Btu per sq ft per day respectively.

Of the 182 days, the percentages having vertically incident radiation greater than 2000, 1500, 1000 and 500 Btu per sq ft per day are shown in Table 2.

TABLE 3—SUMMARY OF ENERGY LOSSES AND GAINS PER SQUARE

TEST CELL	S'VM SOLAR ENERGY TRANSMITTED AND ABSORBED	OUTWARD LOSSES FROM COLLECTOR SURFACE		SOLAR INPUT INTO TEST CELL[a]
	1000 Btu	1000 Btu	% of S'VM	1000 Btu
1	78.4			78.4[b]
		72.6	92.6	5.8
2,3	78.4	55.4	70.6	23.0
4	78.4	63.2	80.6	15.2
5	78.4	65.8	83.9	12.6
6	78.4	64.6	82.4	13.8
7	67.7	51.6	76.2	16.1

[a] Average of two methods of measuring net solar input.

† In Figs. 6-8, the totals for the months are plotted on the 15th day. The connecting lines are for convenience in following trends.

If all of the transmitted and absorbed solar energy had been available to heat the interiors of the test cells, there would have been more than enough to keep all of them except test cell 1 at 70 F (Table 3). A large proportion of the energy was lost outward through the glass, both in the daytime when the outer curtains were up, and at night when they were drawn.

The net solar input was derived by two methods:

1. By finding the difference between the transmitted-and-absorbed solar energy (S'_{VM}) and the calculated outward heat loss (Q_C) from the collector surface.
2. By finding the difference between the total heat loss from the test cell exclusive of window losses (Q_W) and the measured electrical input (Q_E).

These two values should be the same:

$$S'_{VM} - Q_C = Q_W - Q_E \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

Actually, the two methods differed from month to month, as was to be expected of methods which depend upon finding a relatively small difference between two fairly large numbers, as is true of both sides of the equation. The difference was largest in April when the heating requirement was the smallest. For the five-month period the two methods differed by 15.8 percent. Figs. 8A and 8B give the net solar input derived by the two separate methods, and Fig. 8C gives the average of the two. Because the chances of error are about equal in the two methods, the average values are employed in the subsequent discussion.

Fig. 9 is a cumulative plot of the total solar incidence, the proportion transmitted through two glass plates and absorbed by the collector surface, the proportion lost outward, and the net energy available for house heating. Because test cells 2 and 3 had similar net inputs, they are averaged together on Fig. 9 as condition C. Test cell 6 represents condition A, and test cell 4, condition B.

These trends are summarized for the individual test cells in Table 3 which gives the values of solar energy transmitted and absorbed by the collector

FOOT OF COLLECTOR AREA (*December-April, Inclusive*)

TEST CELL	SOLAR INPUT INTO TEST CELL[a]	HEAT REQUIREMENTS		
	% of S'VM	1000 Btu	% of S'VM	Percent Gained from Sun
1	100.0[b]	116.1[b]	148.1[b]	67.5[b]
	7.4	43.5	55.5	13.3
2,3	29.4	61.2	78.0	37.6
4	19.4	32.6	41.6	46.6
5	16.1	32.6	41.6	38.7
6	17.6	33.4	42.6	41.4
7	23.8	33.4	49.4	48.2

[b] Based on total heat loss area (including south window). All other values are based on floor, north wall, and ceiling area only.

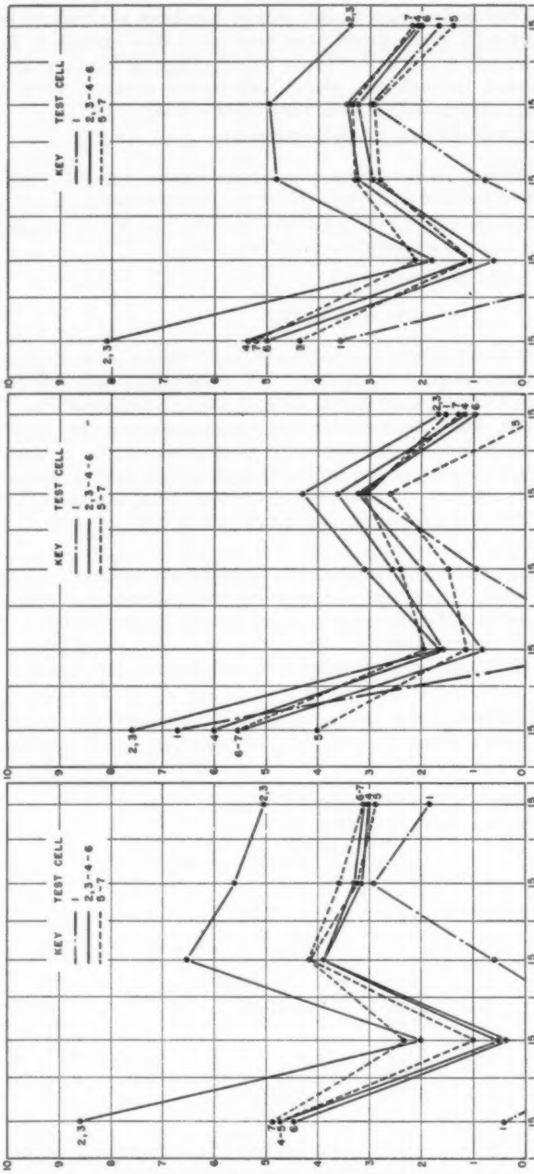

FIG. 8. MONTHLY NET SOLAR ENERGY GAIN OBTAINED BY DIFFERENCE BETWEEN (A) TOTAL HEAT LOSS AND ELECTRICAL INPUT; (B) TRANSMITTED AND ABSORBED VERTICAL SOLAR INPUT AND OUTWARD HEAT LOSSES FROM COLLECTOR SURFACES; (C) AVERAGE OF A AND B

surfaces, the portions of this energy which were lost outward through the curtains and glass, the net gains, the heat requirements, and the fractions of the heat requirements which were supplied by the sun. The net solar input values used in the table are the averages of the two methods employed for their determination. The outward losses are the differences between the solar energy transmitted and absorbed (S'_{VM}), and the net solar inputs into the various test cells. The importance of the outward losses is emphasized by Table 3. Because there was no practical way of eliminating all convection losses around the double curtains, and because the theoretical efficiency of the aluminum coating of these curtains was not attained in a practical installation,

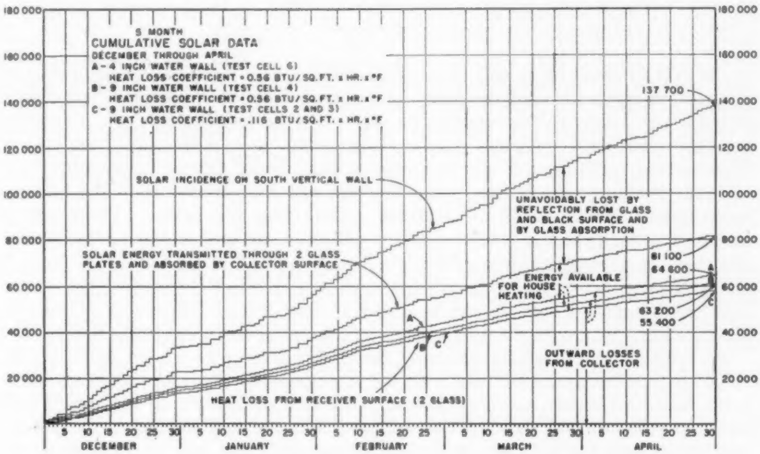

FIG. 9. CUMULATIVE SOLAR DATA FOR FIVE MONTHS DECEMBER-APRIL

the actual overall heat-loss coefficients were high, as shown in Table 1, and the outward heat losses ranged from one and one-half times to two times as high as the theoretical losses. As can be seen from Table 3, the consequence was that on the average less than half the heat requirements of the test cells were supplied by the sun.

A comparison of Tables 1 and 3 shows that if the theoretical heat-loss coefficients had been attained and the outward heat losses had been cut one-third to one-half in test cells 2-7, there would have been sufficient net solar input to make all except possibly test cells 2 and 3 self-sustaining. Because the theoretical heat-loss coefficients could not be attained, the cells all required auxiliary heat. A much better means of preventing outward heat losses must be found than double or triple glass plus double curtain, or the heat-loss coefficients of the floor, north wall, and ceiling must be reduced.

If heat storage were separated from heat collection, so that outward losses from the collector surfaces were confined to the daytime when the collector surfaces were warm but night-time losses were all but eliminated, the efficiency

TABLE 4—COLLECTOR WITH REMOTE STORAGE
(ENERGY GAINS, LOSSES, AND HEAT REQUIREMENTS
CAMBRIDGE, MASS., DECEMBER 1947-APRIL 1948)
(1000 BTU PER SQ FT OF COLLECTOR AREA)

TEST CELL	2-3	4	5	6	7
Transmitted and Absorbed Energy	78.4	78.4	78.4	78.4	67.7
Daytime Losses	30.2	31.8	34.6	33.1	24.5
Night-time Losses	4.5	4.8	5.2	5.0	4.4
Net Gain	43.7	41.8	38.6	40.3	38.8
Total Heat Requirement	61.2	32.6	32.6	33.4	33.4
Percent of Heat Requirements	71.4	128.0	118.0	120.5	116.0

of the system could be greatly improved. Table 4 sets forth the transmitted and absorbed energy; the daytime heat losses; the calculated night-time heat losses through the south wall assuming the same kind of construction and therefore the same heat-loss coefficient as the floor, north wall, and ceiling, and using the observed night-time temperature drops from inside of cell to outdoors; the net gain resulting; the heating requirements; and the percentage of the heating requirements which would be obtained under these conditions. These values assume that daytime temperatures at the collector surfaces would be unchanged and that the heating requirements of test cells would remain the same. Recognizing these approximations, it still appears that in all cells except 2 and 3, the heat requirements would be met by using remote storage instead of storage at the collector. In any event, there could be a marked increase in efficiency.

Test cell 1, the cell which had only the double glass plus double curtain, provides an interesting comparison with the storage-wall cells. As shown in Table 3, the total heat requirements of this cell were 116,100 Btu per sq ft of south wall area, of which 43,500 Btu represented the losses through the floor, north wall, and ceiling, and the balance, the losses through the glass and curtains. Because all of the transmitted and absorbed energy reached the interior of the cell, the

TABLE 5—AVERAGE MONTHLY TEMPER-

t_r = test cell temperature (with mean deviations)

TEST CELL		1	2	3
December (t_o = 29.8)	t_r	74.2 ± 10.6	68.1 ± 1.9	68.1 ± 1.5
	t_c		79.4	79.7
January (t_o = 22.9)	t_r	69.5 ± 6.8	66.4 ± 1.2	66.2 ± 0.9
	t_c		68.3	68.7
February (t_o = 26.6)	t_r	76.8 ± 8.3	71.6 ± 1.0	71.4 ± 0.9
	t_c		79.0	78.9
March (t_o = 38.9)	t_r	76.9 ± 8.5	72.2 ± 1.7	71.6 ± 1.0
	t_c		80.0	80.6
April (t_o = 48.1)	t_r	75.2 ± 6.3	72.4 ± 1.8	71.9 ± 0.9
	t_c		80.2	81.1

net gain to the cell as a whole was 100 percent of this energy figure, or 67.5 percent of the total heat requirements of the cell. The outward losses through glass, however, were 72,600 Btu per sq ft of south glass, leaving a net gain of only 5,800 Btu per sq ft of south glass or 13.3 percent of the requirements of floor, north wall, and ceiling. The total heat requirement was 148.1 percent of the transmitted and absorbed energy, so that even with a double curtain plus double glass of 100 percent efficiency, the total requirements could not have been met.

The solar gain of test cell 1 was obtained mostly by allowing the test cell to overheat. If the temperature had been maintained at a level of 70 F as in the other cells by ventilation or shading, the heat loss of cell 1 exclusive of the south window would have decreased from the overheat condition, 43,500 Btu per sq ft of south window, to 39,500. The 4,000 Btu due to overheat when compared with the 5,800 Btu actually gained shows that a large part of the gain would disappear if the overheating were eliminated. It appears, thus, that the net solar gain of a double glass south wall plus a double aluminum curtain drawn at night is slight, if at all existent. The psychological effect of a large south-facing glass area in winter is generally excellent and this may well be its principal asset.

Temperatures

Further insight into the behavior of solar units is gained from the record of temperatures attained in the storage walls in relation to indoor and outdoor temperatures. Table 5 sets forth the average monthly indoor and outdoor temperatures and the average temperatures attained by the containers for the period December—April. The maximum and minimum values are given in Table 6.

During December and January, which were the periods of the first system of controls, the average cell temperatures were somewhat lower than during the period February—April, when the second system of controls was in operation.

atures and Mean Deviations (F)

t_c = container temperature t_o = outdoor temperature

Test Cell	4	5	6	7
December (t_o = 29.8)	69.1 ± 1.6 83.8	68.9 ± 1.0 89.1	70.8 ± 2.7 85.1	70.8 ± 2.8 89.8
January (t_o = 22.9)	66.5 ± 1.1 68.7	66.7 ± 1.4 70.0	66.8 ± 1.7 70.7	67.0 ± 2.1 73.5
February (t_o = 26.6)	71.6 ± 0.9 80.4	71.7 ± 0.9 83.5	71.7 ± 1.2 81.9	72.1 ± 1.4 86.5
March (t_o = 38.9)	72.3 ± 1.6 82.0	72.4 ± 1.3 85.1	73.3 ± 2.6 83.5	73.7 ± 2.7 87.4
April (t_o = 48.1)	72.8 ± 1.5 82.2	72.5 ± 1.1 86.2	73.8 ± 2.3 82.9	74.1 ± 2.4 85.5

The collector temperatures in cells 2 and 3, which had the heat leaks, were generally lower than in the rest of the cells, although the difference between these two cells and cells 4 and 6 was not great. Collector temperatures in cells 5 and 7 were in general appreciably higher than in the other cells. Cell 7, with triple glass and 4-in. containers attained the highest temperatures. Under January's abnormally poor solar conditions, there was little difference among cells 2 to 6, and even cell 7 was not markedly higher than the others.

Table 6 shows that peak temperatures of approximately 105 F to 110 F were occasionally found in the 9-in. containers, (cells 2 and 5) and that the 4-in. containers (cells 6 and 7) occasionally rose to above 120 F. In spite of these

TABLE 6—MAXIMUM AND MINIMUM WALL TEMPERATURE (F)

ROOM	1	2	3	4	5	6	7
December							
Max.		96.0	97.0	102.0	109.5	118.5	121.0
Min.		62.0	62.0	62.5	63.5	57.5	62.0
January							
Max.		91.0	93.5	93.0	97.5	114.0	116.0
Min.		57.5	57.0	57.5	57.5	56.0	59.0
February							
Max.		98.0	97.5	101.0	104.0	119.5	121.5
Min.		66.5	66.0	66.0	68.5	61.5	65.5
March							
Max.		98.0	97.0	108.5	102.0	122.5	124.5
Min.		65.5	65.0	63.0	65.5	61.0	65.5
April							
Max.		92.0	94.5	94.5	104.0	104.0	104.5
Min.		67.0	67.0	67.5	69.0	63.5	66.5

peaks, the average monthly temperatures in no instance exceeded 90 F (Table 5), and for the 9-in. containers generally ran below 85 F. As was to be expected, the greatest fluctuation between maximum and minimum temperatures occurred in the thinner (4-in.) containers. April showed a marked drop in maximum temperatures because the angle of incidence became large (approximately 55 to 60 deg at noon.)

Although this installation employed water walls, the results are applicable to other heat storage media, because the percentage collection is determined by the temperature of the collector as that affects outward losses, and the average wall temperatures of 80-90 F or less are about as low as are practicable. Consequently, the use of any other wall, either thicker or different, could not produce any marked decrease in outward loss and therefore any marked increase in performance. This is particularly true of Glauber's salts or other fusible materials having melting points in the vicinity of 90 F. The results, then, may be considered applicable to any heat storage medium.

Conclusions

1. The attractive simplicity of the south-facing storage wall, (combining collection, storage, and heating) is offset by the excessive heat losses outward from the collector surface which is warm at all times and therefore loses heat whether collecting heat or not. To make this system feasible, a practicable way must be found of decreasing materially the outward losses during periods when the sun is not shining.

2. The thicker the storage wall, the better its performance, but increasing the thickness 2¼ times as in this experiment (4-in. *vs.* 9-in. water walls) does not increase the efficiency proportionately. Increasing the thickness still more or changing the type of storage medium would have little effect.

3. The greater the heat requirement, the greater the efficiency of the storage system in terms of net solar gain. The two cells with the heat leaks gained 29.4 percent of the transmitted and absorbed solar energy, compared to 19.4 percent for the similar cell without heat leak. The energy gained, however, is a smaller proportion of the total requirement of the more poorly insulated structure as compared with the well-insulated structure.

4. Triple glass in place of double glass increases both the net solar gain and the gained proportion of the total heat requirement. The increase is about 15 percent as evidenced by the 41.4 percent of total heat requirement gained by the 4-in. wall with double glass as compared with the 48.2 percent gained by the similar wall with triple glass. The 4-in. wall with triple glass is approximately the same as the 9-in. wall with double glass.

5. When heat requirements are appreciable, (as in cells 2 and 3) there is little difference between the radiant method and the convection method of transferring heat from the storage wall to the room.

6. Outward losses with the storage wall system range from 70 to 84 percent of the transmitted and absorbed solar energy. The percentages of total heat requirements supplied by the storage walls range from 38 to 48 percent; consequently, more than one-half to six-tenths of the total heat must be supplied by auxiliary sources.

7. If remote storage is substituted for the storage wall approximately 70 to 130 percent of the total heat requirements can be supplied by the sun, but periods of several days of little or no sun (as occurred in January) may deplete the system to the point where auxiliary heat is required in any event.

8. In spite of the relatively low performance of the storage wall, it is better than the no-storage wall. The no-storage wall does show some net gain but it is small compared with the storage wall. The psychological effect, on the other hand, may well be in favor of large south-facing glass areas.

Design Curves

Curves which permit some measure of design of storage walls have been drawn on the basis of the performance of the 4-in. and 9-in. thick storage walls (Fig. 10). In these curves the ordinate is the fractional savings calculated by placing in the numerator net useful solar heat gain during the heating season and in the denominator the total transmitted and absorbed solar energy during the same period. The abscissa is the ratio of the house heating requirements to the transmitted and absorbed solar energy, also during the heating season.

If no auxiliary heat were to be required, the storage wall performance should fall along the diagonal line. The actual performances of the 9-in. walls with double glass for heat loss coefficients of 0.116 and 0.056, and of the 4-in.

wall with triple glass, are plotted, and a curve is drawn. The performance curve of a 4-in. wall is estimated on the basis of the single point for 4-in. wall and double glass and the shape of the curve for 9-in. walls. If, then, a house were to be designed for a ratio of house heating requirement to absorbed solar heat of 0.3, a house requiring no auxiliary heat would require a ratio of net useful solar heat collection to absorbed solar heat of 0.3. The 9-in. wall with double glass would provide 0.225 or 75 percent of the total requirement, leaving 25 percent to be supplied by auxiliary heat. It can be seen that the smaller the ratio of house heating requirement to absorbed solar heat, the more nearly

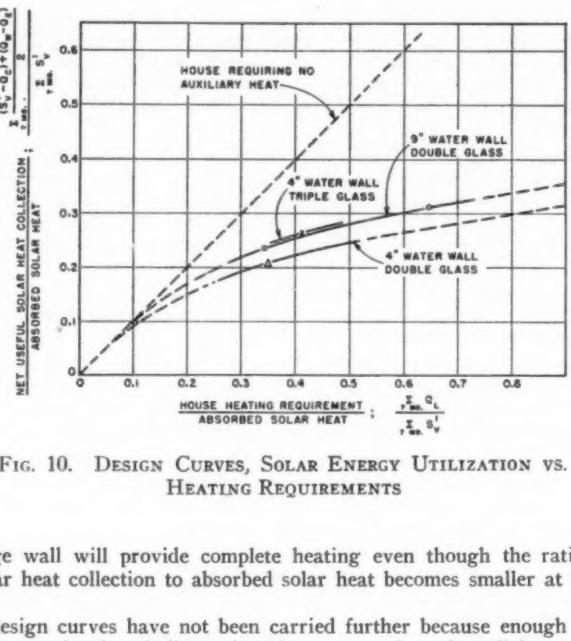

FIG. 10. DESIGN CURVES, SOLAR ENERGY UTILIZATION VS. HEATING REQUIREMENTS

the storage wall will provide complete heating even though the ratio of net useful solar heat collection to absorbed solar heat becomes smaller at the same time.

These design curves have not been carried further because enough evidence has been accumulated to indicate that the system of south wall heat collection should not be employed without modification. The outward heat losses are so large that high efficiency cannot be expected.

If the wall thickness were greatly increased, or if a storage medium of very large heat capacity were employed, no major net gain in heat collection could be expected because a certain wall temperature must be maintained to carry on the mechanism of heat exchange from the storage system to the room. This temperature level fixes the outward heat losses and also limits the heat capacity of the system because the heat capacity must be small enough to allow the solar input to raise the temperature to the necessary level. The conclusion is therefore reached that significant increases in efficiency can be attained only by removing the heat storage function from the heat collecting function, so that excessive outward heat losses can be markedly reduced. In the subsequent work carried

out as a part of this project, the collection and storage functions have been separated.

ACKNOWLEDGMENTS

The authors gratefully acknowledge the assistance and criticism given by several individuals in the preparation of this paper, particularly Prof. H. C. Hottel, chairman of the M.I.T. Committee on Solar Energy Conversion, and Profs. L. B. Anderson and A. L. Hesselschwerdt. The work reported in this paper is part of the continuing project in solar energy conversion established at the Massachusetts Institute of Technology by the gift of Godfrey L. Cabot.

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DISCUSSION

G. V. PARMELEE, Cleveland, Ohio: The authors have presented useful information on a problem that has been of great interest to engineers and the general public for a number of years.

The combination collector, heat storage and space heater described in this paper is a very inexpensive and attractive means of utilizing solar energy. It is disappointing to learn that the losses from the collector surface are so great. However, the large differences in the monthly net solar energy gain as determined by two heat balances described in the paper suggest perhaps that the instrumentation was not sufficiently complete to account accurately for all of the heat losses upon which the performance is based.

In each case, the reported performance of the collector was dependent in part upon the calculated heat losses. In our experience with a solar calorimeter at the A.S.H.V.E.

Research Laboratory, we have found it absolutely necessary to meter accurately the heat flow through various heat leaks of the calorimeter with heat meters or equivalent means in order to get consistent and reliable results.

I would like to ask if the heat storage was calculated from the measured temperature rise of the water in the chambers.

Is any further development contemplated with regard to reducing heat leakage from the storage wall and improving the wall's performance? For example, aluminum foil curtains might have been substituted for the aluminum painted curtains. There is considerable difference in the radiation characteristics of foil as compared to aluminum paint.

I would also like to ask how the transmittance values for the configurated glass were obtained. In some of the work that we have been doing at the Laboratory which has not yet been reported, the maximum transmittance at normal incidence for the configurated glasses we have studied was not over 80 percent as compared to very nearly 90 percent as indicated in the paper.

AUTHOR'S CLOSURE: Mr. Parmelee quite properly points out that there is a discrepancy in the figures for heat loss as determined by the two different methods employed. Such a discrepancy is probably unavoidable, considering the fact that each method involves a small difference between two large quantities. The discrepancy is relatively small compared to the large, accurately measured, electrical heating requirement which indicated that the efficiency of the system was not high, no matter how calculated.

It is quite possible that a double curtain of aluminum foil would have been more efficient than aluminum-painted fabric. Some work was done with foil. The indications, however, were that the convection and infiltration losses around the curtains were the primary reasons why their efficiency was less than expected.

Some practical building and architectural considerations entered into the decision to use remote storage in the house which was subsequently built. In the first place, the necessity for raising and lowering insulating curtains is an awkward inconvenience in any installation. Operation could be made automatic by means of sun switches and motors, but costs of installation would rise accordingly. A serious architectural objection is that the south wall, which affords the best opportunity for large glass areas with their decidedly pleasant aspects in winter, are largely or completely taken over by the opaque heating units. Although it could be argued that large windows in the north wall would provide the same amenities, the opinion seems to be that the south-facing large windows are preferable. Unlike south-facing windows, north-facing windows constitute a source of heat loss with little or no compensating gain from the sun.

All of these factors entered into the decision to employ remote heat storage in the small house which was subsequently built.

1385

No. 1385

REMOVAL OF INTERNAL RADIATION BY COOLING PANELS

By MERL BAKER*, LEXINGTON, KY.

INTERNAL energy sources within a structure, both radiant and convective, may represent a large portion of the total cooling load. Any body within the cooled enclosure at a surface temperature greater than the ambient air temperature is a convective heat source, while any body within the enclosure at a surface temperature greater than the mean radiant temperature of the enclosure surfaces is a radiant source. Therefore, a body may be a convective source, radiant source, or simultaneously, a convective and radiant source depending solely on the temperature relationships.

The rate of energy dissipation by natural convection from a vertical or horizontal surface facing down, varies as the 1.12† power of the temperature difference between the surface and ambient air. In contrast, the dissipation rate by radiation is proportional to the difference of the fourth powers of the absolute temperatures of the emitting surface and mean radiant temperature of the enclosure. Therefore, for a simultaneous radiant and convective source, the most general type, the radiation component becomes of far greater significance as the surface temperature increases. For high temperature sources such as incandescent filaments, the radiation component reaches a magnitude in the order of 90 percent of the total energy dissipation.

An analysis of the mechanisms of heat removal from an enclosure reveals that the radiant and convective components of internal energy sources must be treated separately. The convective component leaves the emitting body and raises the temperature of the surrounding air after overcoming the resistance of the surface film. The action of the radiant component is much more direct in that

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† Panel Heating and Cooling Analysis, by B. F. Raber and F. W. Hutchinson. (John Wiley & Sons, Inc., New York, 1947.)

Presented at the 56th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Dallas, Tex., January 1950.

it is transmitted directly from the surface of the emitting body to the enclosure surfaces.

PANEL PERFORMANCE

Temperature. Any surface which is maintained at a temperature lower than other bodies within a system becomes effectively a cooling panel, and this surface acts as a ***sink*** for radiation striking it. The panel may receive energy directly from a source, indirectly by reflection of source emission from other surfaces, or indirectly by reradiation from other surfaces. The source described previously is not limited to any particular type, but the nature of the source will control the temperature required at the panel surface for effective removal. For an enclosure with all surfaces maintained between 50 and 70 F, the loss by radiation and convection from human occupants will vary approximately as the linear function of the temperature differential. In contrast to operation with low temperature sources, the temperature of the panel becomes nearly insignificant when high temperature sources are involved. The radiant energy received by a 70 F panel is insignificantly different from that received by a 50 deg surface. The possibility of panels operating at relatively high temperatures in removing a lighting load is an outstanding economic factor. For installations in which cooled air is circulated to produce comfort, uncooled water may be used to remove a fraction of the radiant component of the lighting load, or about 70 percent of the electrical input to the filament. For rooms possessing external exposures, the removal of the lighting load is largely effected by the cooling action of the outside walls.

Absorptivity. The absorptivity of all nonmetallic surfaces for long wavelength radiation is generally between 0.9 and 0.98. Therefore a simplified relationship neglecting multiple reflections may be applied to nonmetallic receivers for low temperature radiation with an approximate error not exceeding 4 percent. This fortunate simplification is not valid, however, when the radiation is at short wave lengths, near infrared and shorter, as the absorptivity of some nonmetallic surfaces may become as low as 0.2 for visible radiation. Therefore it is necessary to consider multiple reflections in analyzing the transmission of energy emitted from high temperature sources. The basic radiation equation may be applied if the emissivity factor is known. Since this determination is extremely complex for the case of an enclosure with multiple reflections, it is convenient to make some simplifications. By considering the transfer of radiant energy initially emitted from one equivalent gray area A_1 to another gray area A_2 when the surrounding surface consists of one gray area A_3 which may *see* itself, and when all other surfaces are black, the exact equation† for the emissivity factor for transfer between area A_1 and A_2 is given by

$$S_{1-2} = F_2 F_3 F_4 \left[F_{1-2} + (1 - \alpha_3)\, F_{1-3} F_{3-2} F_1\right] e_1\, \alpha_2 \quad \dots\dots\dots \quad (1)$$

where

$$F_1 = \frac{1}{1 - F_{3-3}\,(1 - \alpha_3)} \quad \dots\dots\dots\dots\dots \quad (1a)$$

† Panel Heating and Cooling Analysis, by B. F. Raber and F. W. Hutchinson. (John Wiley & Sons, Inc., New York, 1947; Chapman and Hall, London, 1947.)

SYMBOLS

α = absorptivity
A = area, square feet
C = conductance of wall, inside surface to outside air, Btu per (hour) (square foot) (Fahrenheit degree)
E = strength of source, Btu per hour
E' = $m\,E$
m = fraction of source dissipation that reaches the surfaces
E_n'' = fraction of energy reaching surface n initially radiated from filament
F = configuration factor for radiant heat exchange between surfaces. Subscripts of F refer to the surfaces by number or by letter. *e.g.* subscript (1-2) refers to exchange from surface 1 to surface 2
h_c = unit conductance for thermal convection Btu per (hour) (square foot) (Fahrenheit degree)
h_r = unit transfer for thermal radiation, Btu per (hour) (square foot panel) (degree difference of MRT and panel)
L = subscript referring to filament
(MRT) = mean radiant temperature, Fahrenheit
n = any surface
Q = quantity of heat transfer, Btu per hour
S_{1-2} = shape factor allowing for multiple reflection in evaluating the radiant transfer from surface 1 to 2
T = absolute temperature, Fahrenheit
t = temperature, Fahrenheit (subscripts: 1,2,3, inside surface wall No. 1,2,3, a = inside air, o = outside air)
W = rate of air exchange (ventilating rate), pounds per hour
x = subscript referring to combination of all surfaces excluding the emitter and receiver

SUBSCRIPTS

1,2,3, etc. refer to numbered walls.
b,c,p,r refer to back of panel, convection, panel and radiation respectively.

$$F_2 = \frac{1}{1 - (1 - \alpha_1)(1 - \alpha_3)\,F_{3-1}F_{1-3}F_1} \quad \text{(1b)}$$

$$F_3 = \frac{1}{1 - (1 - \alpha_2)(1 - \alpha_3)\,F_{3-2}F_{2-3}F_1} \quad \text{(1c)}$$

$$F_4 = \frac{1}{1 - (1 - \alpha_1)(1 - \alpha_2)\,F_2F_3\,[F_{2-1} + (1 - \alpha_3)\,F_{2-3}F_{3-1}F_1]}$$

$$\overline{[F_{1-2} + (1 - \alpha_3)\,F_{1-3}F_{3-2}F_1]} \quad \text{(1d)}$$

and the net radiant heat exchange is

$$Q = S_{1-2}\,0.174\,A_1\left[\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4\right] \quad \text{(2)}$$

Since Equation 1 involves the product of shape factors, an appreciable error will result for large area unless the product is integrated over both areas.

Application of Equation 2 to give the radiant transfer from an incandescent filament to a surface n is:

$$E_n'' = E' \alpha_n F_2 F_3 F_4 [F_{L\text{-}n} + (1 - \alpha_x) F_{L\text{-}x} F_{x\text{-}n} F_1] \quad \ldots\ldots\ldots \quad (3)$$

where subscript L refers to the filament and x to the combination of all surfaces excluding n.

$$F_1 = \frac{1}{1 - (1 - \alpha_x) F_{x\text{-}x}} \quad \ldots\ldots\ldots \quad (3a)$$

$$F_2 = F_4 = 1 \quad \ldots\ldots\ldots \quad (3b)$$

$$F_3 = \frac{1}{1 - (1 - \alpha_n)(1 - \alpha_x) F_{x\text{-}n} F_{n\text{-}x} F_1} \quad \ldots\ldots\ldots \quad (3c)$$

$$E' = mE \quad \ldots\ldots\ldots \quad (3d)$$

where m is less than, or in the limit equal to unity, and E is the strength of the source. Absorptivity α must be used in Equation 3 rather than emissivity. F_2 and F_4 approach unity because of the very small area of the filament relative to surface n.

Only limited data are available for the absorptivity of wall surfaces for high temperature radiation, therefore the accuracy of Equation 3 is dependent on accurate data for the absorptivity. A value of 0.5 may be taken as a representative value for most light colored wall surfaces. Leopold** gives more detailed information.

The illuminating engineer in no case would select surface materials which absorb a high percentage of the visible portion of the spectrum. However, the air conditioning engineer wants the absorptivity of the panel surface to be high for the most efficient operating conditions. A fortunate characteristic of most light colored nonmetallic surfaces is a greater absorptivity for infrared radiation than for luminous. This feature may be enhanced by application of certain heat absorbing paints. The most satisfactory selective combination surface now available reflects about 75 percent of the visible component and 20 percent of the long wave length infrared radiation.

The use of reflectors to increase the shape factor of the cooling surface with respect to the source will increase, theoretically, the transfer rate of the panel. However, the increase may be small or nonexistent in the actual case because of resistance to conduction between the panel surface and the cooling medium. As the shape factor of the receiving surface with respect to the source increases, a greater quantity of energy will be concentrated on the surface. This increase of energy reception will tend to increase the temperature of the surface which will in turn, because of the greater temperature difference, increase the rate of heat transfer to the coolant. Unless the flow rate of coolant is increased, its temperature will rise, thereby reducing the ability to accommodate the increased energy received by the panel. Therefore, an increase in the shape factor will have only a limited increase on the capacity of the panel, unless an adequate increase in coolant flow rate is made.

** The Mechanism of Heat Transfer, Panel Cooling, Heat Storage, by C. S. Leopold. (*Refrigerating Engineering*, Vol. 54, July 1947, pp. 33-41.)

Heat Balance. At equilibrium for a sealed enclosure, the energy received at an inside surface 1 by radiation and convection must exactly equal that transmitted through the wall by conduction. Hence

$$A_1C_1(t_1-t_o) = \Sigma A_n F_{n-1} h_r\,(t_n - t_1) + A_1 h_{c1}\,(t_1 - t_a) + E_1'' \quad . \; . \; . \; . \quad (4)$$

where C is the conductance of the wall between the inside surface and outside air.

Equation 4 does not include multiple reflections among the room walls, but E'' includes multiple reflections for the high temperature radiation.

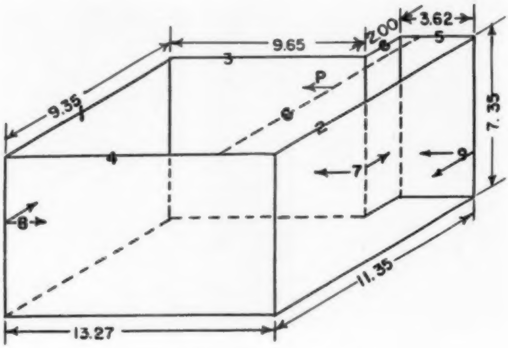

FIG. 1. EXPERIMENTAL ROOM

A solution of Equation 4 gives

$$t_1 = \frac{(F_{1-2}h_{r(1-2)}\,t_2 + F_{1-3}h_{r(1-3)}\,t_3 + \ldots\ldots) + h_{c1}t_a + C_1t_o + E_1''}{(F_{1-2}h_{r(1-2)} + F_{1-3}h_{r(1-3)} + \ldots\ldots) + h_{c1} + C_1} \quad . \; . \; . \quad (5)$$

$$t_2 = \frac{(F_{2-1}h_{r(2-1)}\,t_1 + F_{2-3}h_{r(2-3)}\,t_3 + \ldots\ldots) + h_{c2}\,t_a + C_2t_o + E_2''}{(F_{2-1}h_{r(2-1)} + F_{2-3}h_{r(2-3)} + \ldots\ldots) + h_{c2} + C_2} \quad . \; . \; . \quad (6)$$

$$t_n = \frac{(F_{n-1}h_{r(n-1)}t_1 + F_{n-2}h_{r(n-2)}\,t_2 + \ldots\ldots) + h_{cn}\,t_a + C_n\,t_o + E_n''}{(F_{n-1}h_{r(n-1)} + F_{n-2}h_{r(n-2)} + \ldots\ldots) + h_{c-n} + C_n} \quad . \; . \; . \quad (7)$$

where n is the number of surfaces used, excluding the panel. The additional equation required for a simultaneous solution may be written by an energy balance on the ventilating air. Hence

$$t_a = \frac{h_{c1}t_1A_1 + h_{c2}t_2A_2 + \ldots\ldots + 0.24Wt_o + (E\text{-}E')}{h_{c1}A_1 + h_{c2}A_2 + \ldots\ldots + 0.24W} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad (8)$$

EXPERIMENTAL RESULTS

In order to verify the theoretical equations previously stated, it is necessary to obtain data and information so that a heat balance may be written on the enclosure and the panel. These data include: (1) temperature of room air; (2)

temperature of air behind each enclosure wall; (3) inside surface temperature of each wall; (4) average temperature of panel; (5) panel coolant flow rate; and (6) the change in coolant temperature between panel entrance and exit. In addition, thermal conductance of each wall, combined radiation and convection coefficients for exterior surfaces, convection coefficients for internal surfaces, complete shape factor analysis, surface absorptivity characteristics, and the dissipation rate of the source must be known.

The thermal conductance of each of the enclosure walls was evaluated by application of thermal conductivity data for the particular construction. The combined film coefficients for the outside surfaces were approximated by data

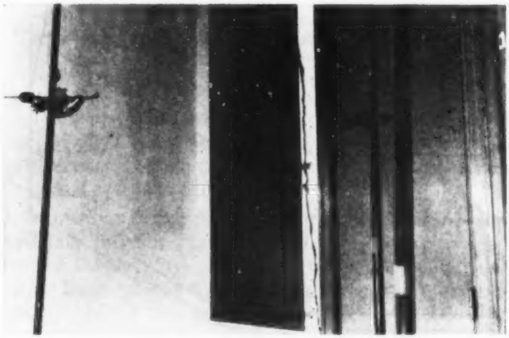

FIG. 2. THE ROOM OUTSET VIEWED FROM THE CENTER OF THE ROOM

given in the A.S.H.V.E. GUIDE, while inside film coefficients for convection from a heated surface to air at room temperature are approximately 1.1, 0.7 and 0.4 for horizontal plates facing up, vertical plates, and horizontal plates facing down respectively. The absorptivity of the white matt surface of the enclosure was taken as 0.5 and 0.93 for high and low temperature radiation respectively. The remaining requisite items were determined by experiment.

Experimental data pertinent to this subject were taken by the author at Purdue University.[△] In order to interpret the data relative to this subject, it is believed necessary to analyze the conditions under which the data were taken.

Experimental Room. The enclosure consists of a sealed room with the overall dimensions being 13.27 by 9.35 by 7.35 ft high with a 3.62 by 2.00 ft outset. The details are shown by Figs. 1 and 2. Fig. 1 also shows the location of the panel, area 105 sq ft. Fig. 2 depicts the outset as viewed from the center of the room. The source considered consisted of an incandescent bulb located approximately one foot below the ceiling at the center of the room.

The interior of the room is finished with plaster characterized by a white painted matt surface. The panel consists of water coils embedded in the ceiling

△ Master's Thesis, Merl Baker, February 1948, Mechanical Engineering Department, Purdue University, Lafayette, Ind.

and covering a major portion of the area. The coil is constructed as a parallel series of one-half inch copper tubes, all connected into parallel headers. Approximately one-half the tube diameter is buried in the plaster.

Temperature Measurement. Twenty-five thermocouples were installed on the panel surface. These couples were copper-constantan located at the centers of rectangles approximately 22 by 26 in., thereby forming an approximate uniform pattern (Figs. 3 and 4). A uniform pattern is necessary if the average panel temperature is to be obtained by the arithmetic average of the point values. To be statistically exact, the rectangles must be replaced by squares, but in this case, the geometry of the surface dictated the use of rectangular sections in preference

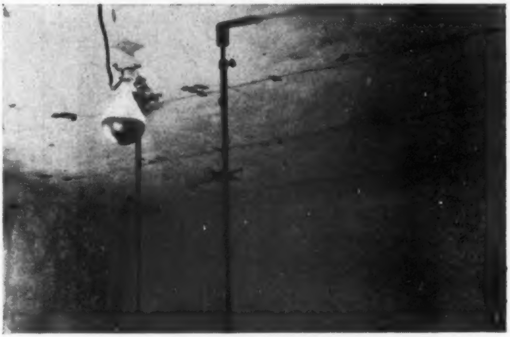

FIG. 3. DISTRIBUTION OF PANEL SURFACE THERMOCOUPLES

to square ones. Since the rectangles used were nearly square, the error may be considered negligible.

Each couple was placed into a prepared groove, the depth being approximately equal to one-half the diameter of the thermocouple junction, and was held in place by translucent tape firmly pressed over the junction. The thermocouple leads were held against the surface for a distance of about six inches. Six thermocouples were installed to measure the room air temperature near the ceiling, floor and mid plane of room. Initially many positions were checked, but it was found that the air for a particular horizontal plane remained essentially constant throughout the room except for positions very near the source, therefore, it was concluded that two vertical positions would be adequate. One vertical stand, located between the source and one interior wall remained fixed while the other one was frequently changed. Negligible fluctuations were observed among corresponding temperatures. The temperature gradient between the ceiling and floor was found to be less severe when a radiant source was employed than with an equally rated convective source. Additional thermocouples were employed to measure the temperature of the air behind each wall. An ice bath was used for the reference junction for all couples.

The calorimetric measurements for the panel coolant appear to be the most critical. The inlet and outlet coolant temperatures were continuously recorded

by a thermocouple potentiometer combination with the flow rate being determined by a rotameter. The flow rate was kept low in order to obtain as great a temperature rise as possible, thereby contributing to the accuracy of thermal measurements. However, low rates were more susceptible to pressure fluctuations and created a greater non-uniformity of temperature across the panel surface.

Source Location. The source was located at the center of the room, off center with respect to the panel, with the center of the filament being one foot

FIG. 4. TWO FOOT SQUARE REFLECTOR AND 1000 W. CLEAR BULB

below the ceiling for a 1000 watt bulb and somewhat less for smaller bulbs. When used, reflectors were supported in a horizontal plane approximately four inches below the filament center.

Four aluminum foil reflectors were employed: (1) one foot square plane; (2) two foot square plane; (3) one foot square plane with a two inch band; and (4) one foot square plane with a four inch band.

Convective Equivalent. The convective equivalent is defined as the hypothetical capacity of a convective source possessing a heat dissipation rate to the air and on to enclosure surfaces just sufficient to maintain the room air temperature at the same level as that resulting from a radiant source. The difference between the source strength and the convective equivalent is the energy portion which is ineffective in raising the room air temperature. Fig. 5 depicts the contrast between a convective and radiant source of the same strength.

Tables 1, 2 and 3 summarize the experimental data.

Confirmation of Equations. Equations 3, 4, 7, 8 may be checked by comparing resulting calculations with experimental data. Test one is calculated by the theoretical equations. Equation 3 is applied to compute the heat transfer to each surface resulting from that initially emitted by the source. Equations 7 and 8 are applied to obtain the surface temperatures of the enclosure. Table 4 gives a list of the calculated surface temperatures.

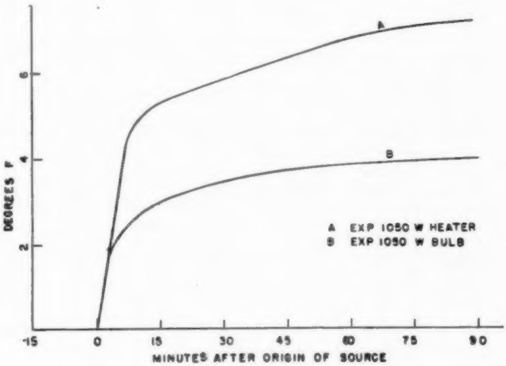

FIG. 5. TEMPERATURE RISE OF ROOM AIR

Heat Balance on Panel. Applying Equation 3, the energy received by the panel surface, directly or by reflection *from the incandescent filament,* is computed as follows:

$$F_1 = 1.67$$
$$F_3 = 1.08$$
$$E_p'' = 0.7 \times 3590 \times 0.5 \times 1.08\,[0.406 + (1 - 0.5)\,0.594 \times 0.192 \times 1.67]$$
$$E_p'' = 678 \text{ Btuh (From incandescent filament)}$$

The energy emitted from bulb glass as low temperature direct radiation may be approximated as follows: A value of 286 Btuh is obtained by assuming that approximately 20 percent of the filament energy is dissipated at or within the globe and by assuming that 40 percent of this quantity is dissipated by radiation. Since the shape factor with respect to the bulb is 0.406, the energy received by the panel from the bulb glass is:

$$0.406 \times 286 = 115 \text{ Btuh (from bulb glass)}$$

The total direct and indirect radiation received by the panel from filament and glass bulb is:

$$Q_L = 678 + 115 = 793 \text{ Btuh}$$

TABLE 1—SUMMARY OF EXPERIMENTAL DATA

Test	Source			Shape Factor	Panel Temperature	Air Temperature		Water Temperature Average	Heat Removed[a] by Panel Coolant
	Rating Watts	Glass Type	Reflector			Midway	Outside		
1	1050	Clear	None	0.406	72.4	79	68	63	1860
2	1050	Clear	*1*	0.615	76.6	82	72	67	1900
3	1050	Clear	*2*	0.726	78.7	82	68	68	1760
4	1050	Clear	*3*	0.718	77.6	82	73	67	2190
5	1050	Clear	*4*	0.827	78.2	81	75	67	1975
1	1050	Clear	None	0.406	72.4	79	68	63	1860
6	1000	Inside Frosted	None	0.406	71.5	80	74	63	1950
7	1000	Silver Base	None	0.850[b]	78.0	81	74	69	1910
1	1050	Clear	None	0.406	72.4	79	68	63	1860
8	500	Clear	None	0.406	68.5	73	61	63	1165
3	1050	Clear	*2*	0.726	78.7	82	68	68	1760
9	1050	Clear	*2*	0.726	87.7	86	65	84	−310
10	1050	Clear	*2*	0.726	99	91	74	100	−750
11	1050	Clear	*2*	0.726	111.8	95	70	119	−1740
12	1050	Clear	*2*	0.726	117.8	99	72	136	−2280

[a] Includes flow to back of panel and gain from storage.
[b] Assumed.
1 One foot square plane.
2 Two foot square plane.
3 One foot square plane 2 inch band.
4 One foot square plane 4 inch band.

TABLE 2—CONVECTIVE EQUIVALENTS

Test	Convective Equivalent Btu per Hour	Percent of Source Input Ineffective in Raising Air Temperature
1	1262	65
2	1275	65
3	1353	63
4	1193	67
5	1248	65
1	1262	65
6	1136	67
7	1140	67
1	1262	65
8	696	59
3	1353	63
9	1849	55[a]
10	1940	62[a]
11	2570	65[a]
12	2645	73[a]

[a] Considering all input of panel goes into heating air.

The energy received by the panel by convection is:

$$q_c = h_c A\,(t_{ac} - t_p)$$
$$q_c = 0.55\dagger\dagger \times 105\,(81.0 - 72.4) = 495 \text{ Btuh}$$

The energy received by the panel by radiation from enclosure is:

$$q_r = A_p h_r\,(\text{MRT} - t_p)$$
$$q_r = A_p h_r\,(F_{p-1}\,\Delta t_1 + F_{p-2}\,\Delta t_2 + \ldots\ldots F_{p-n}\,\Delta t_n)$$
$$q_r = 105 \times 1.05\,[0.187\,(77.54 - 72.40) + 0.308\,(78.58 - 72.40) + 0.137\,(78.07 - 72.40) + 0.066\,(74.81 - 72.40) + 0.307\,(76.60 - 72.40)]$$
$$q_r = 558 \text{ Btuh}$$

The heat lost to coolant from the room:

$$q = Q - q_b$$
$$q_b = AC\,(t_w - t_b)$$
$$q_b = 105 \times 0.119\,(76 - 63) = 163 \text{ Btuh}$$

where

Q = the heat gained by the coolant
q_b = the gain from the back of the panel
$q = 1860 - 163 = 1697$ Btuh

Total heat received by panel	793
	495
	558
	1846 Btuh
Total heat lost by panel (By experiment)	1697
Unbalance	149 Btuh

Room heat balance	
Conduction loss (Calculated)	1595
Panel loss	1697
Radiation loss (Glass window)	145
	3437 Btuh
Input (Electrical input to filament)	3590 Btuh
Unbalance	153 Btuh

The energy balances are remarkably close considering the assumptions made relative to the condition of steady state transfer and in the evaluation of some physical properties.

CONCLUSIONS

Applications of the theoretical equations require the knowledge of the air temperature back of the various walls, the temperature of the air in the enclosure and the conductances of the various walls. These conductances may be obtained by use of tables in the HEATING, VENTILATING, AIR CONDITIONING GUIDE, and air temperatures determined from design data.

Application of Equations 3 and 4 reveals that a room with one-half the surface area exposed to the outside generally possesses enclosure surfaces at tem-

†† Evaluated in accordance with HEATING, VENTILATING, AIR CONDITIONING GUIDE 1948.

peratures low enough to absorb and remove about 50 percent of the total lighting load. This portion of the lighting load is removed from the structure without heating the air; therefore, a consideration that all the electrical lighting load is transmitted to the air may result in an over design of the cooling equipment.

The theoretical equations may also be employed to confirm the results of Table 2, that by the utilization of ceiling cooling panel, the percent of the lighting load directly removed from the conditioned space without heating the air is increased to approximately 65 percent. The calculated heat balance for both the room and panel for test 1 are closely duplicated by the measured quantity.

Experimental data recorded in Table 2 reveal that the percent of filament input (of a 1000 watt clear bulb utilizing a 2 ft plane reflector) which is in-

TABLE 3—WALL CONDUCTANCES INCLUDING BACK (OUTSIDE) SIDE FILM COEFFICIENT

SURFACE		CONDUCTANCE
7	(Combination 3 and 6, Fig. 1)	1.41
8	(Combination 1 and 4, Fig. 1)	0.114
9	(Combination 2 and 5, Fig. 1)	0.675
w	(Window)	0.620
f	(Floor)	0.227
z	(Uncooled ceiling)	0.118

TABLE 4—CALCULATED TEMPERATURES

$t_7 = 77.54$	$t_f = 76.60$
$t_8 = 78.58$	$t_z = 79.70$
$t_9 = 78.07$	$t_p = 72.40$
$t_w = 74.81$	$t_a = 79.00$

effective in heating the room air varies from 63 at a panel temperature of 78.7 F to 73 at a panel temperature of 117.8 F. For the same source and reflector for panel temperatures of 87.7 F, 99 F, and 111.8 F, the percentages are 55, 62, 65 respectively. Because of the possibility of not having steady state transfer, it is believed that the extreme values are not justified and an average value of 65 percent is representative regardless of the panel temperature.

Table 2 also shows that for this particular test the percent of the filament input that is ineffective in heating the air is independent of the type of reflectors used, although the shape factor range considered varied from 0.406 to 0.827. The panel surface temperature varied from 72.4 to 78.2 F, with the higher temperatures increasing the radiant transfer to cooler wall surfaces and thereby compensating for the greater energy received by the panel from the reflectors. However, as previously explained, this may not always be the case.

The type of bulb is not significant as shown by Table 2. The favorable comparison between the calculated and measured panel absorption tends to verify

the hypothesis that multiple reflection must be considered when dealing with high temperature energy sources.

ACKNOWLEDGMENT

Equipment used in obtaining the experimental data for this project was purchased with funds granted by the Copper and Brass Research Association to the Purdue University Engineering Experiment Station.

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4. Cooler Foot Candles for Air Conditioning, by W. G. Darley. (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, pp. 367-378.)

5. Reflection, a Factor in Heating, by M. Fielding, (*Engineer,* Vol. 175, June 18, 1943, pp. 486-487.)

6. Radiating Characteristics of Tungsten, by W. E. Forsythe, E. Q. Adams. (*Optical Society America,* Vol. 35, February, 1945, pp. 108-113.)

7. Influence of Gaseous Radiation in Panel Heating, by F. W. Hutchinson. (A.S.H.V.E. TRANSACTIONS, Vol. 53, 1947, pp. 281-290.)

8. Radiant Heating and Cooling, by C. O. Mackey. (Cornell University *Engineering Experiment Station Bulletin 33,* August, 1944.)

9. A.S.H.V.E. RESEARCH REPORT No. 1192—Panel Heating and Cooling Performance Studies, by B. F. Raber and F. W. Hutchinson. (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, pp. 35-54.)

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11. Lighting and Air Conditioning, by H. M. Sharp. (*Heating and Ventilating,* Vol. 39, November, 1942, pp. 36-46.)

12. Effects of Artificial Lighting on Air Conditioning, by Walter Sturrock. (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, pp. 213, 244.)

13. Radiation and Convection from Surfaces in Various Positions, by G. B. Wilkes and C. M. F. Peterson. (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 513.)

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15. Scientific Basis of Illuminating Engineering; (Parry Moon, New York, London, McGraw-Hill Book Co., 1936.)

DISCUSSION

W. E. LONG, Austin, Tex.: Professor Baker has opened up a new line of thought in our all too frequent method of *guesstimation* relative to the cooling load of air conditioned spaces. The practice of adding from 10 to 20 percent as a *contingency* will continue to be necessary until more concrete facts, such as are presented in this paper, are available to the engineer in the field.

Since I am of the opinion that THE GUIDE should lean heavily toward simplification of data useful to the practicing engineer, I shall not deal with the mathematical development involved in the work done by Professor Baker. The comparison of

theoretical and test results seems to prove that correct assumptions for shape factor, absorptivity of surfaces, and other constants were made. The real test of the value of the work now seems to lie in the compilation of tables or curves from which the field engineer may decide how to evaluate the lighting load in each specific case.

I cannot envision an extensive use of water cooled panels installed for the removal of lighting load. However, it seems that the information presented in this paper might cause us to change our thinking relative to the allowance to be made for such load; certainly this is true if only 35 to 45 percent of the energy input to a light is effective in heating the room air.

C. S. LEOPOLD, Philadelphia, Pa.: The manufacturers' published data on energy distribution of a filament lamp are shown in Fig. 1.

84 percent appears in the form of radiation. Radiation is obviously not heat, but can elevate the temperature of a solid object and, to some extent, can elevate the temperature of a gas, thus appearing in the form of sensible heat. Since transfer by radiation is proportional to the difference in the fourth power of the absolute temperatures, and since the filament operates at approximately 4800 F, it is apparent that the radiant transfer is relatively independent of panel temperature under the temperatures at which we normally live.

Mr. Baker has presented a mathematical analysis of the transfer, and though I do not agree with some of the assumptions used in the mathematical solution, such as using the same shape factor for a coil filament as for an enclosing globe, and the use of the gray body concept, I agree with the general assumption that by multiple reflection substantially all of the radiant output will result in the elevation of the surface temperature of the enclosing room and will either be conducted into the structure or will heat the air by convection. The panels have a flowing coolant appreciably below the radiating source (4800 F); hence the heat can be eventually removed by the coolant so that the over-all mathematical result can be predicted just by knowing the energy distribution of the lamp.

An internal heat load having a low radiant component ultimately dissipates its energy output in heat of the walls. Transmission in this case is by convection. It is to be noted that in either of these cases the room air temperature will rise appreciably, and heat balances are usually made for a substantially constant room air temperature and for summer conditioning on the assumption that the outdoor temperature is greater than the indoor temperature. I therefore believe that the conclusions are somewhat optimistic. Where there are windows, previous investigators have shown a portion of the radiation from the lamp can find its way out through the window glass.

I believe that the problem can be more readily visualized by separating the radiant components into those directly absorbed by the ceiling, regardless of the ceiling temperature within ordinary limits, and the radiation absorbed by a cool ceiling due to the difference in temperature between ceiling and other room surfaces.

In my paper of reference, I made such an analysis experimentally. The result is shown in Fig. 2 where IRT means independent radiant transfer, and is the percentage of the total energy input of the lamp which is absorbed by the ceiling, substantially independent of the ceiling temperature.

In a subsequent paper on the design and use of a hydraulic analogue, I have indicated that the time lag in reaching equilibrium is of long duration, and in the discussion of Professor Mackey's paper* on the mathematical analysis, I presented data showing that in the eleven hour-a-day operation of conventional cooling systems with internal load applied for nine hours, the ordinarily calculated cooling requirements were not realized in full even at the end of one week's operation. The storage effects are of the utmost importance.

* Heat Gains Are Not Cooling Loads, by C. O. Mackey (A.S.H.V.E. TRANSACTIONS, Vol. 55, 1949, page 413).

Professor Long questioned the applicability of panel cooling. For over two years I have had in operation an experimental installation located in a New York City office building. Four rooms, three with south exposure and one interior, are panel cooled. Supply air to these rooms is limited to that required for freshness and dehumidification. Approximately 60 percent of the total internal sensible cooling is accomplished by panels. The operation of panels with the solar load is similar to their operation with luminaires. The actual amount of high temperature radiation entering the room depends, in part, on the means of shading the glass.

C. E. BENTLEY, San Francisco, Calif.: We feel that there is great possibility in the use of cooling panels in the following instances:

1. Reduction in size of air handling apparatus for a given air supply temperature split.

2. Reduction in required air supply temperature split in critical comfort occupancy areas.

3. Reduction in size of compressor plant where design wet bulb is low enough to make practical the use of a cooling tower.

There are few metropolitan localities in which it would be feasible to waste the panel coolant water and thus effect a saving in installed refrigeration. If refrigeration compressor plant size were reduced, and a separate cooling tower used, the lowest practicable initial coolant temperature would be roughly 7 F above the design, wet bulb—82 F, for 75 F, wet bulb.

Most installations in high occupancy areas on the West Coast are using tubular lamps which require about one-half the wattage for the equal amount of illumination as incandescent; therefore, many times do not cause the excessive high heat build-up. Also, these fixtures have reflectors directing light and radiant heat away from the ceiling.

We feel that each installation must be analyzed from all bases of design and economy before a definite recommendation can be made for the installation of cooling panels and realize that this article would be a great help for this study.

AUTHOR'S CLOSURE: First, I would like to comment on Professor Long's suggestion that the equations are rather involved. I am of the opinion that no panel heating or cooling job can be accurately calculated without the use of more or less involved equations. This probably is of greatest significance when cooling panels are to be designed to remove the lighting load.

However, it would be advantageous, I am sure, to the practical engineer if some simplifications could be made. An endeavor should certainly be made to simplify the equations but without too great a decrease in accuracy.

In connection with Mr. Leopold's disagreement with the assumption made in the calculations, I may add that a gray body concept was not assumed in general. The problem was divided into two phases. First, the gray body concept was assumed for the case of the low temperature surface to surface radiation. This is generally accepted. It is realized that the wall surfaces are selective absorbers if a large portion of the radiation spectrum is considered. This is why a value of 0.5 was used for the absorptivity of the wall surfaces when irradiated by high temperature energy contrasted to a value of 0.9 for the case of low temperature exchange.

It would be more nearly correct if the selective absorber concept were taken further. However, to achieve a practical development we have two choices in this particular case: (1) consideration of multiple reflections or (2) a more complete consideration of selective absorptivity. It is believed that the change of absorptivity after successive reflections is relatively unimportant from the standpoint of a rational design procedure. However, it is recognized that multiple reflections must be considered in

deriving an accurate procedure. This was done in the analysis presented. A consideration of the change of absorptivity after each successive reflection would be extremely complex. It is doubtful that such a procedure could be practical.

The geometric shape factors from the incandescent filament to the surfaces were evaluated by considering the filament a point source. This would not be exactly true for points near the filament, but is accurate for areas a significant distance away. It may be noticed that for this particular test, 145 Btu per hr was lost through the window glass by radiation.

I am in agreement with Mr. Bentley's statements on the favorable use of the cooling panels.

1386

No. 1386

BASEBOARD RADIATION PERFORMANCE IN OCCUPIED DWELLINGS

By G. S. MacLeod* and C. E. Eves**, Chicago, Ill.

Previous papers by Kratz, Harris and Weigel[1, 2] concerning performance of baseboard radiation have reported laboratory and test house findings. It was felt that the next logical step in furthering the knowledge of this branch of the heating art would be a study of baseboard radiation performance in a variety of occupied dwellings. The purpose of this paper is to present test data gathered during such a study.

In conducting these tests, it was considered important that any household activities which might influence temperature or humidity conditions should be carried on at the pleasure of the occupants.

Test Program

Five houses of varied size, shape, and construction were chosen as subjects for this study. In view of the fact that the occupants of these houses were under no obligation to the sponsors of the project, several considerations influenced the planning of the program and the choice of instrumentation. The more important of these factors were:

1. No damage to decorations or furniture would be permissible.
2. No permanent test equipment could be installed.
3. All apparatus must be readily transportable.
4. A minimum of interference with occupants' normal activities would be allowable.
5. One week in each house was considered the maximum time in which to gather data.

The general program for each house was as follows:

First day. Set up instruments, made fuel-consumption and combustion-efficiency tests, drew floor plans, calculated house heat loss, and started automatic recording instruments.

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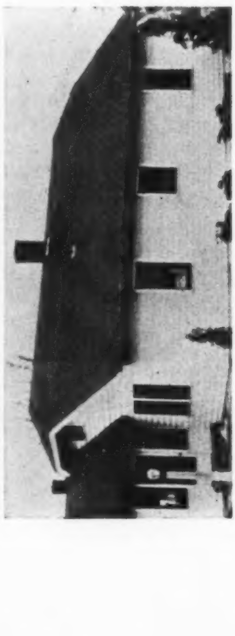
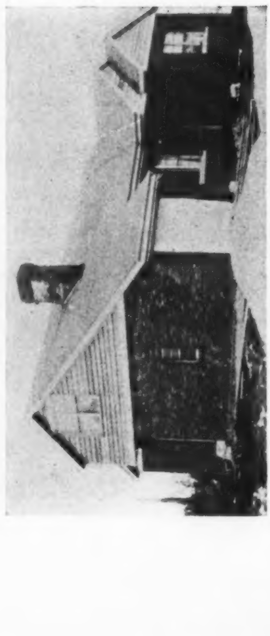
* Test Engineer, Sears, Roebuck and Co. Member of A.S.H.V.E.

** Former Test Engineer, Sears, Roebuck and Co.

Presented at the 56th Annual Meeting of The American Society of Heating and Ventilating Engineers, Dallas, Texas, January, 1950.





CASE No. 1 *(upper left)*
CASE No. 2 *(center)*
CASE No. 3 *(lower left)*
CASE No. 4 *(upper right)*
CASE No. 5 *(lower right)*
(See Legend on Opposite Page)

FIG. 1. EXTERIORS OF THE FIVE DWELLINGS
(CASES NOS. 1 TO 5)

LEGEND FOR FIG. 1

Case No. 1—Southwest view. *Walls:* frame with wood siding and sheathing exterior; painted plaster board interior. *Ceiling:* flat, built-up tar and gravel on wood sheathing; vented above insulation; painted plaster board interior. *Floor:* wood floors on joists over basement in north half of house; asphalt tile on concrete over gravel fill in south section of house. *Insulation:* 3⅝ in. rockwool bats in ceiling; 1 in. cotton bats with reflective surface in all exterior walls. *Boiler Location:* in basement below northeast bedroom.

Case No. 2—Northeast view. *Walls:* frame with wood siding and sheathing exterior; lath and plaster interior. *Ceiling*: lath and plaster, unfloored attic above. *Floor:* wood floors on joists over full basement. *Insulation:* 3⅝ in. rockwool bats between joists in ceiling; loose rockwool in exterior walls. *Boiler Location*: in basement below bathroom.

Case No. 3—Northwest view. *Walls:* first floor, brick veneer with plaster on plaster board interior; second floor, frame. *Ceiling:* plaster on plaster board interior, unfloored attic above west half of first floor; ceiling of second floor, plaster on plaster board under rafters. *Floor:* wood floor on joists over full basement. *Insulation: 2* in. vegetable fiber bats in all exposed walls and in ceiling over west half of first floor. *Boiler Location:* in basement below garage.

Case No. 4—Southwest view. *Walls:* frame, brick veneer, or stone veneer exterior; lath and plaster interior. *Ceiling:* lath and plaster under partly floored attic. *Floor*: wood floor on joists over full basement. *Insulation:* 3⅝ in. rockwool bats in all exterior walls and between ceiling joists. *Boiler Location:* in basement below playroom.

Case No. 5—East view. *Walls:* brick veneer exterior; 1 in. knotty-pine panel or lath and plaster interior. *Ceiling:* lath and plaster under flat, "built-up" roof. *Floor*: terrazzo on joists over half-basement. South wing of structure over a three-foot crawl space. *Insulation:* 6 in. loose mineral wool between ceiling joists only. *Boiler Location:* in basement below west bedroom.

Second, third, and fourth days. Took four complete cycles of manually-recorded data each day. Cycles were started at 7:00 a.m., 11:00 a.m., 3:00 p.m., and 7:00 p.m.

Fifth day. Dismantled and packed apparatus, and obtained general comments from the occupants regarding their reactions to, and their experiences with, the baseboard heating system.

TEST EQUIPMENT

Houses. The five houses chosen for this project represented a fair cross-section of the types of North American single dwellings in which baseboard radiation might logically be installed. The number of rooms heated ranged from five to fourteen. The number of occupants ranged from two to six. Three of the houses were heated entirely by means of baseboard radiation. The other two had small amounts of standing radiation. Three of the systems were oil-fired, one was gas-fired, and one was coal, stoker-fired. Figs. 1, 2, and 3, and Table 1 present more detailed information.

Radiation. The radiator sections used in this study were of hollow steel, shaped to resemble wood baseboard. The nominal length of each section was

FIG. 2. FLOOR PLANS OF DWELLINGS STUDIED

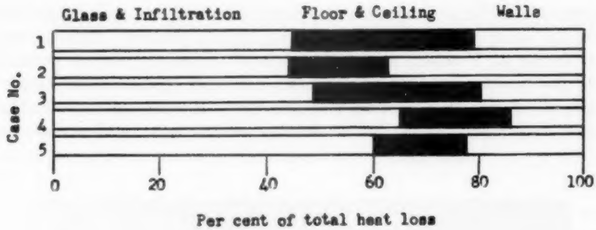


Fig. 3 Percent of Total Heat Loss

three feet; the height of the front face was about nine inches. No provisions were made for convective air currents between the radiator sections and the walls on which they were mounted. As may be seen in Fig. 2, the radiation was installed on exposed walls where conditions would permit. Forced hot water was the heating medium used in all of the five systems.

Instruments and Controls Used in Test Program

Controls. Operation of the circulating pump in four of the systems was controlled by a snap-action room thermostat. An immersion thermostat controlled the firing device so that the boiler water temperature was maintained between 180 and 200 deg.

In the other system, the circulating pump operated continuously, and the oil-burner was controlled by the room thermostat.

Table 1—House Details

1. Case Number	1	2	3	4	5
2. No. heated rooms	5	6	10	14	13
3. Stories	1	1	1½	1½	1
4. Floor area, sq ft	954	937	1531	2413	2133
5. Heated volume, cu ft	7518	7678	11,836	18,669	17,770
6. Glass & door area, sq ft	212	158	352	613	504
7. Total heat loss (80 F diff.)	42,300	35,000	92,700	91,100	113,600
8. Percent total heat loss:					
a. Walls	20.8	37.8	19.7	13.7	22.0
b. Ceiling	18.0	17.1	27.3	19.2	5.8
c. Floor	16.2	1.1	4.3	1.8	11.5
d. Infiltration	19.5	17.0	14.5	20.2	20.9
e. Glass & doors	25.5	27.0	34.2	45.1	39.8
9. Ceiling height, in.	97	102	96	96	100
10. Insulation	Full	Partial	Partial (Not completed)	Full	Partial
11. Fuel used	Oil	Gas	Coal (Stoker)	Oil	Oil
12. Basement	½ Exc.-½ Con. Slab	Full Exc.	Full Exc.	Full Exc.	½ Exc.-½ Crawl Space

Instrumentation, Recording. A recording potentiometer and copper-constantan thermocouples were used to obtain continuous temperature data of the following: ice bath for 32 F reference; outside air in the shade; south vertical outside surface of the building; south vertical sol-air temperature of a black surface; room ambient of the thermostat; living room radiator surface, outlet; living room radiator surface, inlet; boiler inlet pipe surface; and boiler outlet pipe surface.

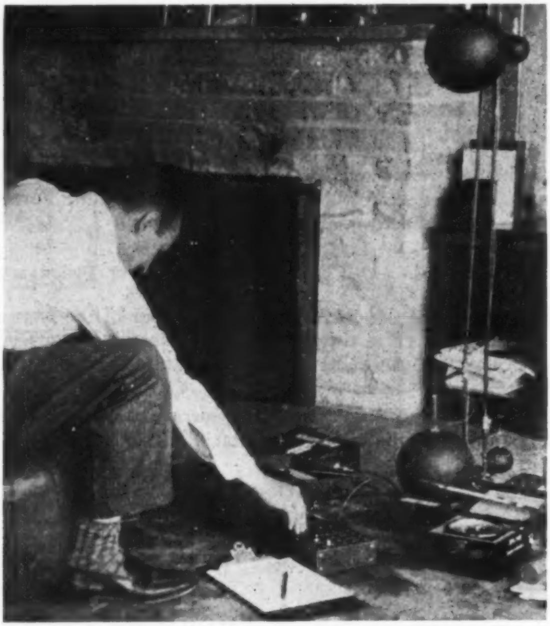


FIG. 4. INDOOR TEST APPARATUS

The operating periods of the circulating pump, and the periods and total operating time of the firing device, were recorded during the entire test period at each house.

Instrumentation, Indicating. Combustion efficiency and fuel consumption determinations were made on each installation before the actual test cycles were begun. No adjustments were made to firing device during the balance of the test.

A portable potentiometer and test stands (see Fig. 4) were used in determining the room air temperature and the radiation-convection temperature. Four No. 30 iron-constantan thermocouples were suspended from each stand, two at the 3-in. level and two at the 48-in. level. One of these at each level was placed in the center of a copper globe, which had been painted with two coats of a

standard flat back. Two previous papers,[3, 4] describe the details and use of globe thermometers. The other thermocouple at each level was suspended in the air about one inch from the periphery of its nearest globe. Lead wires from the four thermocouples on each stand terminated in an eight-contact plug near the base of the stand. One of these test stands was placed midway between the intersection of the living room diagonals and each corner. In each of the other heated rooms one stand was placed as near the center of the room as furniture would permit.

When the 3 in. diameter globes and the 8 in. diameter globes were suspended within a few inches of each other, so that neither one shaded the other from the main source of radiant heat, the observed globe temperature of the 8 in. was of the order of 1½ deg higher than that of the 3 in. globe. Thus, the mean radiant temperature (MRT) of the 8 in. globe was also higher under the same conditions. However, for convenience in transporting and to inconvenience the occupants less, the smaller size globes were used for most observations. Air temperature variations, both horizontally and vertically, were so small that an unshielded single thermocouple was considered sufficiently accurate for the purposes of this project.

Air temperatures at the 3-in.-below-ceiling level were taken directly above each test stand by means of a thermocouple taped 3 in. from the end of a long rod.

A selector-switch box and 20-ft lead wire extensions, having appropriate plugs and sockets, made it possible to connect and disconnect lead wires quickly and to make many temperature observations without moving the switch box or potentiometer. Also, the hazard of occupants tripping over wires was reduced to a minimum.

A thermo-anemometer was used to determine the velocity of the room air at a point within 2 in. of each black globe. An air-velocity traverse was also taken in the living room of each house in the four vertical planes established by the test stands, 30 in. above the floor, at the following points: 3 in. and 12 in. from the outside wall, and 12 in. and 3 in. from the inside wall.

The wet and dry bulb temperatures at the center of the living room were observed each period by means of a sling psychrometer. Atmospheric pressure readings were obtained from an aneroid barometer. The direction and velocity of the wind were determined with a velometer and compass. Weather data such as rain, snow and cloudiness were also observed and recorded.

DISCUSSION OF RESULTS

Winter Comfort Conditions. A thorough discussion of what constitutes comfort in a heated structure is outside the scope of this paper. However, at least the measurable, physical factors that influence the sensation of comfort must be considered in an evaluation of heating system components.

Published papers pertaining to winter comfort indicate that among the more important of these influencing factors are the following: temperature of the air; air velocity; mean radiant temperature; and relative humidity.

In the broad sense, winter comfort conditions actually imply a state of equilibrium between the heat losses of the human body and the heat gains. Consequently, maintaining winter comfort resolves itself into the control of the *net*

effect of as many of the influencing factors as possible, and over as much of the occupied area of the structure as possible.

All occupants of the houses studied agreed that they knew of no other type of heating system that maintained such uniform, comfortable conditions in a dwelling. These human reactions, though they do not lend themselves to scien-

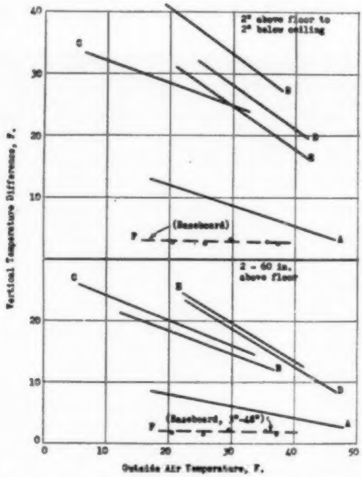


FIG. 5. AVERAGE VERTICAL TEMPERATURE DIFFERENCES OBSERVED IN FIVE BASEBOARD-HEATED DWELLINGS. DATA SUPERIMPOSED ON A REPRODUCTION OF FIG. 10 OF BUREAU OF STANDARDS REPORT BMS-108

(Reproduced by permission of the National Bureau of Standards.)

LEGEND FOR FIG. 5

A = Gravity hot water heating system with radiators.
B = Electric heater with gravity circulation through a plenum chamber.
C = Electric heater with forced circulation (780 cfm) through plenum chamber.
D = Oil furnace with forced circulation through plenum chamber.
E = Gas space heater with a disk fan.
F = Forced hot water using baseboard radiators.

tific measurement and evaluation, are an indication of the achievement of the result hoped for, *i.e.,* the control of the *net effect* mentioned previously. Perhaps the ensuing discussion of the individual influencing factors will explain the occupants' reactions.

DISCUSSION OF TEMPERATURES

Air Temperature Distribution. Dr. E. U. Condon, director of the National Bureau of Standards, makes the following comment in a Bureau publication[5],

"Uniformity of temperature throughout houses is a tacitly accepted American ideal of heating, ...". As that publication deals almost entirely with air-temperature variations in a test bungalow somewhat similar from a heating standpoint to the first two houses studied in this baseboard radiation project, the data contained therein were used as a basis for comparing the baseboard performance.

Fig. 5 shows that the average air temperature differences from floor to ceiling in the baseboard-heated homes were appreciably less than those in the Bureau of Standards test house when using other types of heating systems.

In a few instances during the baseboard radiation tests, the air temperature near the ceiling was observed to be lower than that near the floor. In the summary of test data (Table 2) these cases are referred to as *temperature inversions.* This phenomenon would probably not occur in structures heated with the more conventional systems. However, as floor-to-ceiling differentials approach zero, and as the convection effect decreases, radiant heat from the sun coming through large glass areas would tend to raise the temperature of the floor and rugs, and indirectly the air temperature near the floor, without a corresponding increase in temperature of the air near the ceiling.

These temperature inversions were not included in the averages of air temperature differences as it was believed that doing so would influence the averages unfairly.

The 3 deg floor-to-ceiling difference observed in this project agrees substantially with the results of baseboard radiation tests at the University of Illinois as reported by Kratz and Harris[1], and by Weigel and Harris[2].

It was also observed that in Case No. 4 in the three rooms heated with free-standing, thin-tube radiators, the vertical air temperature differences were in the 9-to-11 deg range.

The variation in temperature from room to room at various levels was small, considering the fact that the water flow rate to individual radiators had not been finally adjusted in any case by means of balancing valves. Most horizontal differences observed were between two and four degrees when no unusual conditions prevailed.

These near-zero differentials occurred under diverse conditions of floor construction and glass area.

The satisfactory heating of rooms with concrete slab floors agrees with the observations of Weigel and Harris[2] at the University of Illinois who state, *the radiant baseboard is particularly adapted to maintaining comfortable floor slab temperatures in a basementless structure.*

Room Air Velocity. In the 1949 edition of the HEATING, VENTILATING, AIR CONDITIONING GUIDE, p. 783, the following statement is made, *velocities less than 15 fpm generally cause a feeling of air stagnation, whereas velocities higher than 65 fpm . . . . . may result in a sensation of draft.* It will be noted from the summary of test data (see Table 2) that the averages of velocities at the room centers were between 17 and 24 fpm at the 3 in. level, and between 9 and 14 fpm at the 48 in. level. However, no user complained of any stagnant air conditions. Velocity observations at points directly above the baseboard radiators and at other points in the rooms showed only slight departures from the values at the room centers.

TABLE 2—SUMMARY OF TEST DATA

CASE NO.	1	2	3	4	5
Test date, 1949	2/8-2/11	2/15-2/17	2/22-2/25	3/1-3/4	3/8-3/11
Average vertical temperature difference, F:					
3 in. above floor to 48 in. above floor	1.9	1.4	2.2	2.3[a]	1.5
3 in. above floor to 3 in. below ceiling	2.9	2.4	3.0	3.3[a]	2.4
Average of the maximum horizontal air temperature differences:					
3 in. above floor	6.1	3.7	4.0	6.4	4.5
48 in. above floor	4.9	3.1	5.4	6.2	3.9
3 in. below ceiling	4.2	3.4	—	—	—
Average air velocity, all rooms, fpm					
3 in. above floor	21.8	21.0	17.7	23.2	23.0
48 in. above floor	9.5	12.7	12.7	12.0	13.4
Maximum air velocity, fpm					
3 in. above floor	70	44	45	110[b]	60
48 in. above floor	25	21	27	22	26
Average relative humidity (living room), percent	37	43	47	37	46
Average outdoor temperature in shade, F	20.5	25.5	35.2	29.6	37.0
Average indoor temperature (at thermostat), F	73.0	73.0	70.0	71.0	71.0
Indoor-outdoor temperature difference, F	52.5	47.5	34.8	50.4	34.0
Average heat loss during test period, Btu/hr	27,800	20,800	40,300	57,400	48,300

(Continued on Opposite Page)

TABLE 2—SUMMARY OF TEST DATA *(Continued)*

	3 in.-48 in.	3 in. flr-3 in. clg.	3 in.-48 in.	3 in. flr-3 in. clg.	3 in.-48 in.	3 in. flr-3 in. clg.	3 in.-48 in.	3 in. flr-3 in. clg.	3 in.-48 in.	3 in. flr-3 in. clg.
Test averages of vertical temperature differences, F:										
Living room, N.E., area	4.3	6.2	1.4	2.0	2.7	3.8	1.8	2.5	1.7	3.4
Living room, S.E. area	2.6	3.4	1.8	1.9	2.7	3.8	1.8	2.4	2.0	3.4
Living room, S.W. area	1.1	3.0	0.9	1.2	3.9	4.8	2.3	3.3	2.0	2.7
Living room, N.W. area	4.1	4.7	1.3	1.0	2.6	3.4	1.9	3.0	1.7	2.5
Living room, average of 4 areas	3.2	4.3	1.3	1.7	3.0	3.9	1.9	2.8	1.9	3.0
Dining room	—	—	—	—	2.6	3.5	4.3	6.3[d]	1.9	3.0
Kitchen	2.0	3.2	1.0	2.1	3.2	3.7[e]	4.4	9.1[c]	2.3	4.5[e]
Bathroom	0.8	1.7	1.0	2.1	1.7	2.9	5.1	11.3[c]	2.0	2.9
Northeast bedroom	1.6	2.7	1.5	1.8	0.8	1.2	2.1	3.4	1.2	2.0
Northwest bedroom	1.9	2.6	0.8	1.5	—	—	—	—	1.8	2.4
Southeast bedroom	—	—	—	—	3.9	4.9	—	—	0.7	1.3
Southwest bedroom	—	—	—	—	2.5	3.6	—	—	1.4	1.9
No. of vertical temperature inversions observed[f]	15	4	9	5	2	1	2	5	5	4
Percent of total difference observations	15	4	13	7	2	1	2	5	4	3
Average amount of inversion, F	1.4	1.1	0.6	0.7	0.2	0.3	3.3	0.8	0.6	0.7

[a] includes baseboard-heated rooms only.
[b] near open fireplace.
[c] heated with standing radiation.
[d] opens into room heated by standing radiation.
[e] baseboard radiators behind cabinets.
[f] temperature inversion: when air temp. at floor is greater than at higher levels.

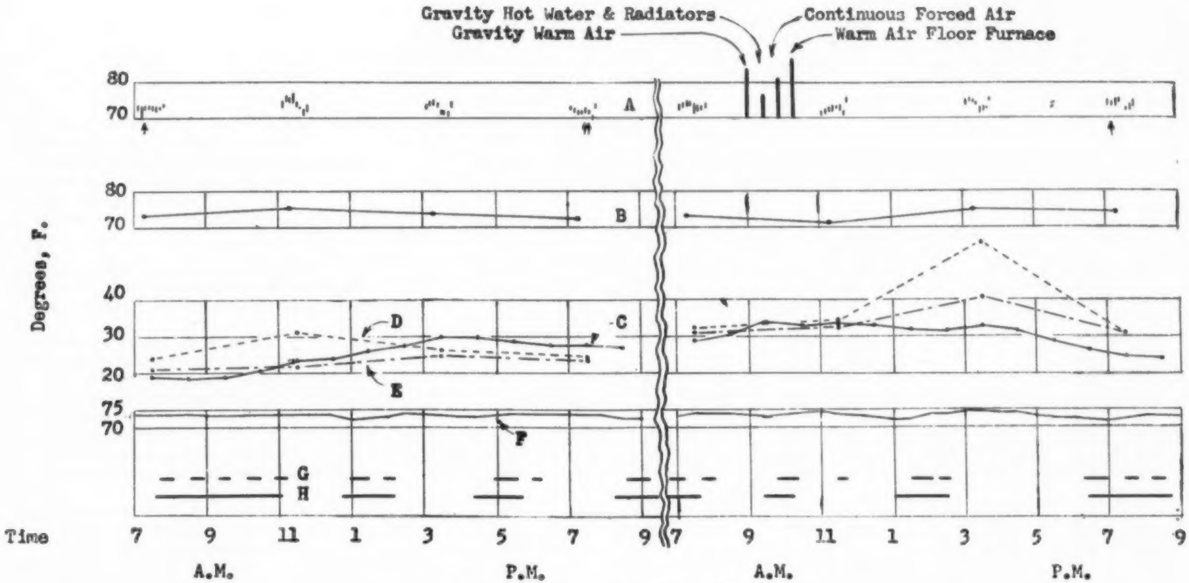


FIG. 6. TYPICAL LOG OF TEST DATA, CASE NO. 2

A. Short lines in Chart A represent room air temperatures and differences between floor and 48 in. levels. The lower end of each line represents ambient temperature 3 in. above the floor; upper end shows temperature 48 in. above the floor. (Exceptions: arrows indicate *temperature inversions* where 48 in. temperature is lower than that near the floor.)
The four heavy vertical lines represent the average vertical air-temperature differences (at 32 F, outdoor) from 3 in. to 60 in. above the floor in the U. S. Bureau of Standards Test Bungalow, using a variety of heat distributing systems. (From Bureau of Standards Report No. BMS-108, page 13.)
B. Mean radiant temperature of the living room, at room center, 48 in. from floor.
C. Temperature at Chicago Airport, U. S. Weather Bureau Station.
D. Sol-air temperature of a black surface, south vertical exposure.
E. Outdoor air temperature in the shade as observed near test house.
F. Air temperature at thermostat.
G. Burner operating periods.
H. Circulator operating periods.

Mean Radiant Temperature. The mean radiant temperature (MRT) of an environment has been defined in a previous A.S.H.V.E. paper[3] as the ***temperature of a uniform enclosure with which one's body would exchange the same amount of energy by radiation as in the actual environment.*** Professor Mackey[4] shows that a small globe thermometer, 48 in. above the floor and at the center of a room, ***gives closely, the mean radiant temperature influencing radiant heat transfer from the standing human body.*** The majority of globe temperatures observed in the baseboard-heated rooms were within one degree of the temperature of the surrounding air.

The MRT of structures heated by conventional systems generally decreases with a decrease in outside temperature. Thus, air temperatures in a room must be increased on cold days and decreased on mild days to achieve the same sensation of warmth. However, in spite of a wide variety of outside temperatures, nearly constant mean radiant temperature and air velocity conditions were encountered in the houses studied. As a result, control of the air temperature alone by means of an ordinary room thermostat satisfactorily governed the comfort conditions during the heating season. This was significant in that it indicated that no complex temperature control apparatus was required for this type of heating system.

Relative Humidity. Averages of the living room relative humidity observations for the five houses ranged from 37 to 47 percent. Nearly all of the individual observations, when plotted against outdoor air temperatures were above the average of 3050 observations reported by Phillips[6] of the Bureau of Standards in a survey of conditions in 215 residences in the Northern United States.

SOLAR RADIATION AND TEMPERATURE DISTRIBUTION

Effect of Solar Heat Gain. Bright sunlight shining through an unshaded glass area would naturally tend to raise the temperature of most objects in its path. The effect of solar radiation was noticed mainly on the globe thermometers, and to a lesser extent on the air temperature. When an individual is comfortable in a shaded portion of a structure, then moves into a position where direct sunlight reaches his body, he will probably have an uncomfortable sensation of warmth. Obviously then, sunlight may be expected to have an effect on comfort conditions in localized areas, regardless of the heating system. However, no general effect of solar heat on the temperature distribution within a house was observed during this series of tests.

Self-Balancing Effect. Case No. 3 presented a number of interesting conditions that would indicate the satisfactory operation of baseboard systems under the most adverse circumstances. This house, though occupied was still in the process of construction. A tarpaulin covered the west entrance to the garage. The north wall of the living room consisted of plaster on plaster board, and was exposed to the cold garage. There was free circulation of air from the garage to the uninsulated space above the living room, the southwest bedroom, and part of the bathroom. Thus, under the conditions prevailing at the time of the test, the living room and the southwest bedroom had an inadequate amount of radiation.

In spite of this lack of balance, both the vertical and horizontal temperature differences compared favorably with the other houses where the conditions were

more nearly in balance. This would lead to the conclusion that when there is free communication between rooms within a structure, and the *total* amount of baseboard radiation is sufficient to offset the *total* heat loss of the house, satisfactory temperature distribution can be achieved, even when the amount of radiation in some rooms is unequal to the heat loss of the corresponding rooms.

CONCLUSIONS

The test results indicated the following general conclusions:

1. Size, shape, or construction materials of the structures had little effect on the overall performance of the baseboard radiation.
2. Air temperature differentials from floor to ceiling and from room to room were less than in houses heated by more conventional systems.
3. Baseboard radiation systems were free from inherent drafts.
4. Indoor relative humidity was observed to be satisfactory without the use of humidification devices.
5. Highly satisfactory results were obtained from the use of simple control systems.

The highly satisfactory comfort conditions reported by the occupants of baseboard-radiation heated residences were generally substantiated by the physical measurements of the thermal environment.

It is hoped that the work on this field project will contribute to the overall knowledge of the heating art, and that those readers interested in pursuing the subject will draw upon the experience of the authors.

The conclusions offered are neither final nor complete, but do indicate that baseboard radiation presents some interesting phenomena which justify further inquiry.

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DISCUSSION

W. A. DANIELSON, Memphis, Tenn.: I would like to ask the authors if they observed any dust streaking from baseboard heating because that is an important thing from the housewife's standpoint.

E. K. CAMPBELL, Kansas City, Mo.: There are some things which are generally overlooked about steam radiators and radiator heating. I want to bring out the

thought that a large part of the results of radiator heating is due to, or by means of, air movement. There are many high ceiling auditoriums, heated with direct radiators where the air movement is so slow that it takes many hours to get the heat down to the floor, even though there is sufficient radiation theoretically to supply the heat load of the room. That explanation of the trouble has not been recognized generally by the steam trade.

In this presentation the matter of draft is spoken of, but drafts are a matter of temperature and not of motion. You do not feel warm air in motion if it is near the temperature of the body but, if the air on the floor is still cold then the motion is very noticeable. Consequently if you have drafts across the floor, and horizontal drafts are all that count, more movement, that is more cold air drawn off, means that you will increase the temperature of the air flowing across the floor and reduce what we term draft.

Large volume air movement, whether it be from a radiator or a fan is the cure for unpleasant drafts, for cold floors or any trouble of that sort. The result is, or the conclusion must necessarily follow, therefore, that the difference between your baseboard radiator and your standing radiator is in the amount of air moved in proportion to the heat that the radiator actually puts out. A long low radiator of a certain number of feet or certain rated capacity will move more air than a standing radiator that stands 32 or 38 in. high where the air has to move up the whole length of it.

Several years ago in an A.S.H.V.E. meeting, some tests of radiators were presented; and I said then that it was not a question of the type of radiator involved, but any large radiator would create more air movement, and therefore, give a better result if only some way could be found to control it and shut off the heat when you had enough.

I want to bring out the fact that the difference between different types of radiators is largely a difference in the amount of air moved.

P. R. ACHENBACH, Washington, D. C.: The authors are to be congratulated on this very understandable paper on the performance of baseboard radiation in the field. It is gratifying that the results obtained in these five occupied dwellings agree generally with the results that were obtained in the laboratory at the National Bureau of Standards on a baseboard convector system.

It is noted, however, that the vertical temperature differences between levels observed for the baseboard convectors tested at the National Bureau of Standards were approximately 150 percent of those reported in this paper for baseboard radiators. There are two reasons that may account for this difference: *first,* with the major part of the heat being emitted by convection from the baseboard convectors it is expected that a greater vertical temperature difference would result than for the baseboard radiators for which an appreciable part of the heat was emitted by radiation and; *second,* there was no heat emitted in the basement of the house used for our laboratory tests whereas the heating plant was located in the basement in all five houses discussed in this report. Our observations have been that heat liberated by piping or boilers beneath the floor is very useful in raising the temperature near the floor surface above.

I was interested in the fact that the two globe thermometers, 3 in. and 8 in. in diameter, respectively, indicated temperatures differing by about 1½ F. I would like to ask the authors why they preferred the 3 in. globe and which they considered the better instrument for determining the mean radiant temperature.

M. K. FAHNESTOCK, Urbana, Ill.: As mentioned in the paper a considerable amount of work has been done at the University of Illinois on various types of residential heating systems. We are always interested in additional data on the subject.

This paper covers a good field survey on a type of residential heating system and presents information useful to the heating profession and to the public. However because of the incorrect inferences which can be drawn, I question the advisability of taking field survey data and comparing it with laboratory test data such as has been done in Fig. 5. If I am not mistaken, the data shown in Fig. 5, comparing the performance of warm air heating with radiant baseboard heating, are data obtained in a test bungalow which is a very small house located inside of a refrigerated space. Am I correct, Mr. Achenbach, or Mr. MacLeod?

MR. ACHENBACH: The data shown was obtained before the enclosure was put around the bungalow.

MR. FAHNESTOCK: Then the test bungalow was exposed to variations in outdoor environmental factors and not to a fixed temperature as in a refrigerated space.

G. S. MACLEOD, Chicago, Ill.: That is right.

MR. FAHNESTOCK: I am glad that this was the case because our experience has shown that results obtained in a room which is surrounded by a refrigerated space maintained at a constant temperature may be quite different from those which are obtained in a house exposed to variable outdoor weather variations. Floor to 60 in. level and floor to ceiling air temperature gradients are usually larger for steady state conditions than for variable conditions.

I personally do not believe that the floor to ceiling air temperature differences shown in Fig. 5 are representative of those obtained with conventional forced circulation warm air heating systems. I do not know whether they are supposed to be representative or not, but presented as they are on a comparison basis with air gradients obtained with an up-to-date forced circulation hot water heating system they will be so interpreted by most readers. This is unfortunate because I believe the comparison is unfair. I agree that the results which Mr. MacLeod has presented are generally representative of baseboard radiation performance, but I do not believe that those shown in Fig. 5 are representative for warm air systems.

There is another matter which I would like to mention in connection with evaluating the comfort conditions produced in a heated space, whether it be in a house, in the field or in the laboratory. Ceiling and floor surface temperatures should be observed, along with the attic, basement and/or crawl-space temperatures. Basement and crawl-space temperatures, in addition to affecting floor surface temperatures also affect floor to ceiling air temperature gradients.

Another thing which is important in using air temperature gradients as a measure of the performance of automatically fired heating systems is to take complete gradient readings near the end of the on-periods and near the end of the off-periods. What happens in between is important too, but the conditions may differ at different parts of the cycle. When comparing one automatically fired system with another it is well to compare the conditions existing over complete cycles.

There is nothing mysterious about the performance of heating systems using baseboard radiation. The excellent air temperature gradients obtained with them is largely the result of placing the heat into the exposed outside part of the structure at a low level by means of long thin heating elements. Here it counteracts the cold surfaces and makes up the heat losses at the points where they occur.

F. E. INCE, St. Louis, Mo.: I would like to point out that Professor Queer advocated the reduction of relative humidity to prevent the moisture accumulation in walls. Now, I see how you could increase the relative humidity by putting in baseboard radiation. Perhaps this relative humidity is the result of the lower mean temperature in the space, and otherwise perhaps due to the fact that you have low velocity.

With the introduction of single glass windows, I am wondering if this higher relative humidity would also introduce a problem on condensation at the windows. Also, what was the direction and velocity of the air near the windows? Do you have any figures on that?

AUTHORS' CLOSURE: General Danielson asked a question about dust streaking. I might tie that in with the question by Mr. Viessman as to the type of radiation, the type of panel. This was what is known by some manufacturers as the type R, perhaps mistakenly called strictly radiant or entirely radiant. We know there must be some convection effect, that is only reasonable to assume; but this type R certainly—you might put it this way—does a beautiful job of distributing the dirt.

Those of you who have had field experience in the heating business know that you do not eliminate dirt with any particular type of heating system. I expect some reaction from that too, but we certainly have found that baseboard radiation does a very nice job of distributing it. To answer the question specifically, no definite dirt pattern has been observed yet; and some of these installations have been in about three full heating seasons.

I might mention that some of them are on fairly dark walls where there would never be a dust pattern anyhow; but others are on light walls, and certainly the streaking is at the very minimum. We are certainly going to get some patterns with this type of system, but this, frankly, is less dirt pattern than I have ever seen in any type of heating system.

In answer to Mr. Campbell in his discourse on drafts, currents, and convection currents, about all I can say in answer to that is that we reported the things pretty much as we found them with what we considered to be a reliable instrument after having it carefully calibrated with a gasometer and wind tunnel: and we feel that our air velocity figures and directions were reasonably accurate.

I must call to your attention and much of this discussion hinges on the fact that this was a field program where we had to go into an occupied dwelling that belonged to somebody else. We had no control over the house itself; the occupants had no obligation to us. Out of the kindness of their hearts, they allowed us to go in and live with them for practically one full week, from about 7:00 a.m. to 9:00 p.m.

We, therefore, felt that it would certainly be betraying their hospitality and imposing upon them to install any permanent test equipment, that is to actually affix thermocouples to a wall or ceiling or floor. I might even mention that in measuring the surface temperature of the pipes and of the inlet and outlet to the radiants, we did have to tape on our thermocouples. We thought it wiser to give in just a little on accuracy and not to get into any trouble with the owners of the test houses.

Mr. Achenbach mentioned the heating systems being in the basements, and consequently, helping out the floor conditions.

Take Case No. 1. You will recall the small bungalow with the flat roof. It had a half basement and half concrete slab. This might be significant. The living room, the kitchen, and the dinette were on a concrete slab; and yet their two year old and their one year old youngsters were playing on the asphalt tile covered floor, in perfect comfort. Frankly, we sat on the floor to take our observations because the floor was more steady than any table we could find for the potentiometer, and we had a very definite sense of comfort. There certainly was no sense of draft in the unexcavated portion. Also, the floor to ceiling differential in air temperature was essentially the same in the excavated portion as the unexcavated. We feel that that has some significance.

Professor Fahnestock, I believe Mr. Achenbach answered one of your questions. When we are comparing, perhaps we are taking some liberties; but we are working with the best types of comparisons we had at hand. We are not apologizing for it, we are merely facing the fact.

We did not have steady state conditions in our observations nor, I believe, in the Bureau of Standards' as reported in their various heating systems; and in their variety of heating systems they also had transient conditions.

Concerning floor and ceiling temperatures, we did not feel at liberty to affix thermocouples to the plaster. We thought our observations sufficiently close to give us the answers we were after.

Mr. Ince mentioned relative humidity. As I said previously, we do not attempt to make any real scientific explanation of our rather high relative humidities. We did get some condensation, of course, on our single windows. Some of the windows were storm-sashed, some were single. We got some condensation on both of them at low outside temperatures. There again, that was not too objectionable. At least, we did not hear any complaints.

The question of velocity of the air near the windows is a hard one to answer because it did not seem to form any consistent pattern. It was so low that even our very presence would influence the instrument. It is a very sensitive instrument. Velocities in the order of 2, 3, 5 and 10 fpm are so negligible that direction makes apparently no difference.

1387

No. 1387

LABORATORY STUDIES ON HEAT FLOW WITHIN A CONCRETE PANEL†

By C. M. HUMPHREYS*, H. B. NOTTAGE**, C. V. FRANKS††, R. G. HUEBSCHER‡, L. F. SCHUTRUM°, AND D. W. LOCKLIN^Δ, CLEVELAND, OHIO

This paper is the result of research carried on by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at its Research Laboratory located at 7218 Euclid Ave., Cleveland 3, Ohio.

IN THE latter part of 1947, the Society undertook a long-range program of research in panel heating and cooling *to develop data to permit the proper calculation and design of panel heating and cooling systems, with present emphasis on applications for human comfort.* The Technical Advisory Committee on Panel Heating and Cooling, which was formed to assist the Committee on Research and the Laboratory staff in directing the program and interpreting the results, divided the investigation into four divisions, three of which were assigned to separate groups.

Under Group A, which was concerned with heat distribution within and behind the panel, the design and construction of test equipment was begun early in 1948, and laboratory experiments on concrete panels were carried on from the Fall of 1948 to the Summer of 1949.

This paper is a summary of Research Laboratory studies up to the Fall of 1949, including test results, use of electrical analogue, and an interpretation of results in relation to fundamental theory of heat flow within a panel.

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† Abstract of this report presented at the 56th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Dallas, Tex., January 1950.

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Three concrete panels were constructed and tested under conditions to be described. An electrical analogue was developed and used to verify test results, as well as to extend test data to ranges beyond test conditions. Comparisons were then made between the values obtained from tests, those found by the electrical analogue, and those predicted by use of the fundamental theory of heat flow for uniformly spaced pipes buried within a solid panel.

HEAT FLOW TESTS

Three concrete slabs, each 6 ft square and containing 17 tubes spaced on 4-in. centers, were constructed for these studies. The report presented in January, 1949[1] gave details of the method of slab construction and contained a brief illustrated description of the test methods and instrumentation used. Table 1 summarizes pertinent data on the three slabs.

The tubes in each slab were arranged in a manifolded grid so that tube spacings of 4, 8, 12 and 16 inches could be obtained in tests through the use of selective valving. Special heat absorbers and guards, designed and built at the Research Laboratory, made it possible both to measure and to control the rate of heat transfer from each side of the slab. Supply water to the slab and each absorber was independently controlled and could be held to within $\pm$ 0.5 deg through a range of 85 F to 160 F for the slabs and 50 F to 120 F for the absorbers. A sectional view of the slab and absorbers is shown in Fig. 1. The heat flow rate was measured by heat flow meters on each absorber and was checked by calorimetry from measurements of the flow rate and temperature rise of the water passing through each absorber. Fifty-five (55) thermocouples, located

NOMENCLATURE

b = centerline bury of pipe, inches or feet, taken as b_1 or b_2 in accord with Fig. 4.

C = thermal conductance, Btu per (hour) (square foot) (Fahrenheit degree).

C_e = electrical conductance, (reciprocal of resistance in ohms).

D = pipe outer diameter, inches or feet.

E = electrical potential, volts.

I = electrical current, amperes.

k = thermal conductivity, Btu per (hour) (square foot) (Fahrenheit degree per foot).

k_e = electrical conductivity, (reciprocal of resistance in ohms/foot).

L = slab thickness, inches or feet.

m = pipe heat flow rate (source strength), Btu per (hour) (linear foot).

q/A = heat flow rate, Btu per (hour) (square foot).

R_1 = thermal resistance of insulation over slab surface (hour) (square foot) (Fahrenheit degrees) per (Btu).

s = pipe centerline spacing, inches or feet.

t_2, t_1 = mean slab surface temperatures, Fahrenheit degrees.

t_i = mean insulation surface temperature, Fahrenheit degrees.

θ = temperature difference, Fahrenheit degrees.

θ_p = source component of difference in temperature between the mean slab surface and outer pipe surface temperatures, Fahrenheit degrees.

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[1] Exponent numerals refer to References.

within the slab and on the two surfaces, permitted accurate determinations of the temperature patterns. Thermal equilibrium was established for each test condition prior to the collection of test data.

At least five tests were made on each slab with both sides bare, for each of four tube spacings. Conditions for each series of five tests were chosen to give

TABLE 1—SLAB DIMENSIONS

SLAB NO.	NOMINAL TUBE DIAM. IN.	TYPE OF TUBES	DEPTH OF COVER, IN.[a]		NOMINAL SLAB THICKNESS IN.
			Top	Bottom	
1	¾	non-ferrous	3½	5	9⅜
2	¾	non-ferrous	1½	2½	4⅞
3	¾	ferrous	1½	3½	6

[a] Cover depth measured from slab surface to edge (not center) of tubes.

heat releases from the upper surface of the slab of approximately 0, 25, 50, 75 and 100 percent of the heat supplied, the remainder being given up by the lower surface.

Tests were also made at all four tube spacings with the upper surfaces of slabs 1 and 2 covered with carpet and carpet padding, and also with several thicknesses of insulation. Additional tests were made on slab No. 1 after a ⅛-in. asphalt tile floor had been applied to the upper surface.

USE OF ELECTRICAL ANALOGUE

While tests on the third slab were still in progress, an electrical analogue was constructed to verify and extend the range of the thermal data obtained on the slabs. The arrangement of the analogue is shown diagrammatically in Fig. 2 and the parts are shown in Fig. 3.

In using the analogue the following equivalents were employed:

Thermal Quantity	*Electrical Quantity*
Temperature difference, θ	Voltage difference, E
Heat flow rate, q/A	Current, I
Thermal conductivity, k	Electrical conductivity, k_e
Thermal conductance, C	Electrical conductance, C_e

The slab dimensions, and the major thermal quantities involved are shown in Fig. 4.

Referring to Fig. 2, the analogue tank width was 16 in., the same as the maximum tube spacing tested. The slab simulated in Fig. 2 had thickness L, depths of bury b_1 and b_2, and had tubes of diameter D spaced $s = 16$ in. on centers. To simulate the same slab with 8-in. tube spacings, the central electrode would have consisted of two tubes 8-in. apart. Thus the geometry of the slab is completely duplicated between the glass barriers of the analogue.

The air space conductances were simulated by adjusting the glass barriers shown in Fig. 2 so that the ratio of the electrical conductivity, k_e, of the elec-

trolyte to the electrical conductance, C_e, of the glass barrier was equal to the ratio of the thermal conductivity, k, of the concrete to the thermal conductance, C, of the air space, or

$$\frac{k}{C} = \frac{k_e}{C_e}$$

The air space conductances set up in the analogue were related as given to those determined in tests on the slabs. The end electrodes represented the absorber plates of the slab thermal tests.

The use of electric analogue methods in panel studies is also described by Kayan[2] in reporting on an A.S.H.V.E. cooperative research project at Columbia University.

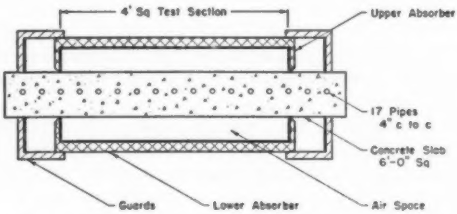

FIG. 1. SECTIONAL VIEW OF CONCRETE PANEL AND ABSORBERS

THERMAL PERFORMANCE DATA

Consistency of the laboratory data was checked as each run was made by plotting the observed performance in a simple but convenient form. The curves of Fig. 5 were drawn through points obtained for bare slabs. Points for insulated slab tests fell on or were very close to the same curves.

The *flux factor* employed as the ordinate on these curves is merely the ratio: (observed rate of heat flow from the pipe to the slab surface concerned) ÷ (rate of heat flow which would exist to this same slab surface if the pipes were replaced by a plane sheet through their centerlines having the same temperature as the pipe surface, the slab surface temperature remaining unchanged). *The flux factor curves show performance trends only; they are not for design use,* being restricted to particular test conditions.

The validity of each performance test was checked by working out the heat balance on the absorber assembly.

INTERPRETATION OF RESULTS

As has already been noted a careful study of the theory of heat flow within a slab was made concurrently with the laboratory testing program. A satisfactory analytical solution was developed, and is presented in another paper[3]. Both

the thermal test data and the analogue extensions thereof were analyzed on the basis of this theory.

Variables Involved and Their Reduction

The heat flow rate through a slab surface, even when limited to steady-state conditions, may vary over a very wide range as influenced by the following

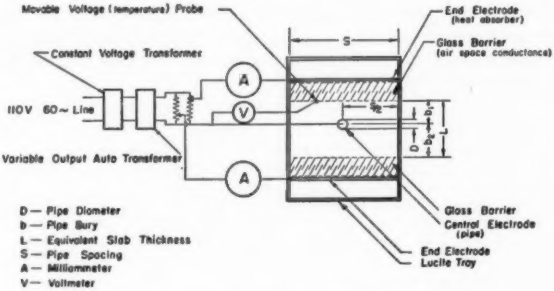

FIG. 2. SCHEMATIC DIAGRAM OF ANALOGUE

variables of construction and operation: (1) environmental conditions (which are represented in these tests by absorber temperatures), (2) water temperature, (3) pipe spacing, (4) pipe bury, (5) pipe diameter, (6) slab thickness, (7) pipe material, (8) insulation (of variable thickness and thermal conductivity)

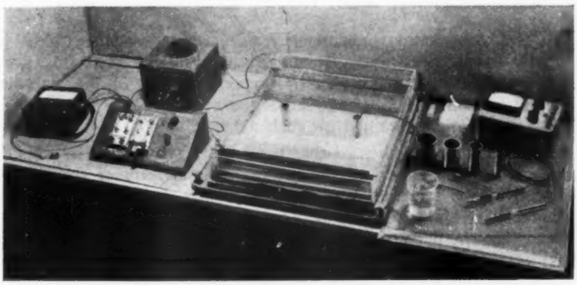

FIG. 3. VIEW OF ELECTRICAL ANALOGUE

on slab surface, (9) slab thermal conductivity, and (10) thermal conductances between the slab surfaces and the respective environments.

Since it was necessary, in organizing the analysis of the data, to simplify wherever possible, the following steps were taken and justified as noted in following paragraphs:

1. Detailed study was limited to the heat flow process within the slab. Absorber temperatures and air-space conductances were replaced as variables of analytical in-

terest by the mean slab surface temperatures. Though it was recognized that the slab surface temperature wave had an influence on the thermal system, it was also realized that the influence was quantitatively small within the scope of the present tests and that an adequate experimental treatment would be rather lengthy.

2. The pipe material (ferrous or non-ferrous) was ruled out as an important *thermal* variable for the concrete slabs involved. Calculations showed that the thermal resistance of the pipe, regardless of the metal used, was entirely negligible in relation to the other thermal resistances involved. The heat flow was therefore analyzed in terms of the pipe surface temperature rather than the water temperature.

3. The four geometrical variables, pipe spacing s, pipe centerline bury b, pipe outer diameter D, and slab thickness L, were reduced to three by considering only the dimensionless ratios s/L, b/L, and D/L.

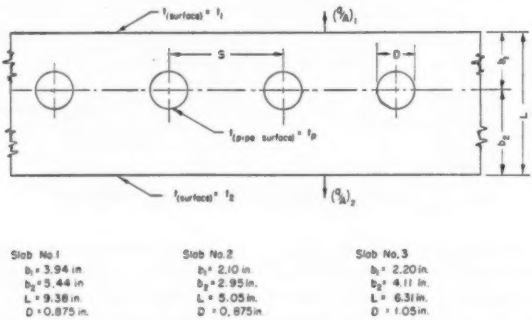

FIG. 4. SLAB THERMAL AND GEOMETRIC QUANTITIES

4. Insulation was considered to be merely a thermal resistance added externally and independently to the heat-flow system within the slab. That this was justified, for the test range involved, was shown by an examination of the slab temperature patterns with and without insulation, by analogue observations, and by the adequacy of the final data correlation obtained on such a basis.

5. The fundamental theory indicated that the results (as plotted in Figs. 7, 8 and 9 discussed later) were independent of the thermal conductivity of the slab. This was also demonstrated by the fact that the analogue results were not dependent upon the electrolyte conductivity. This is true because of the particular method of plotting; it does not mean that the performance of a system installed in the field is independent of the thermal conductivity of the materials used in the panel construction.

Heat Flow and Temperature Difference Components

Interpretation of the experimental data was simplified by following the method of the fundamental theory[3] and resolving the heat flow rate and temperature difference across each slab surface into two components, which may be defined as follows:

1. The *source component* is the magnitude which would prevail if the slab surfaces were isothermal at the same temperature and the tubes were the only sources of heat.

2. The *linear component* is the magnitude which would prevail if the sources (tubes) were removed and the slab surfaces were isothermal but at different temperatures. In

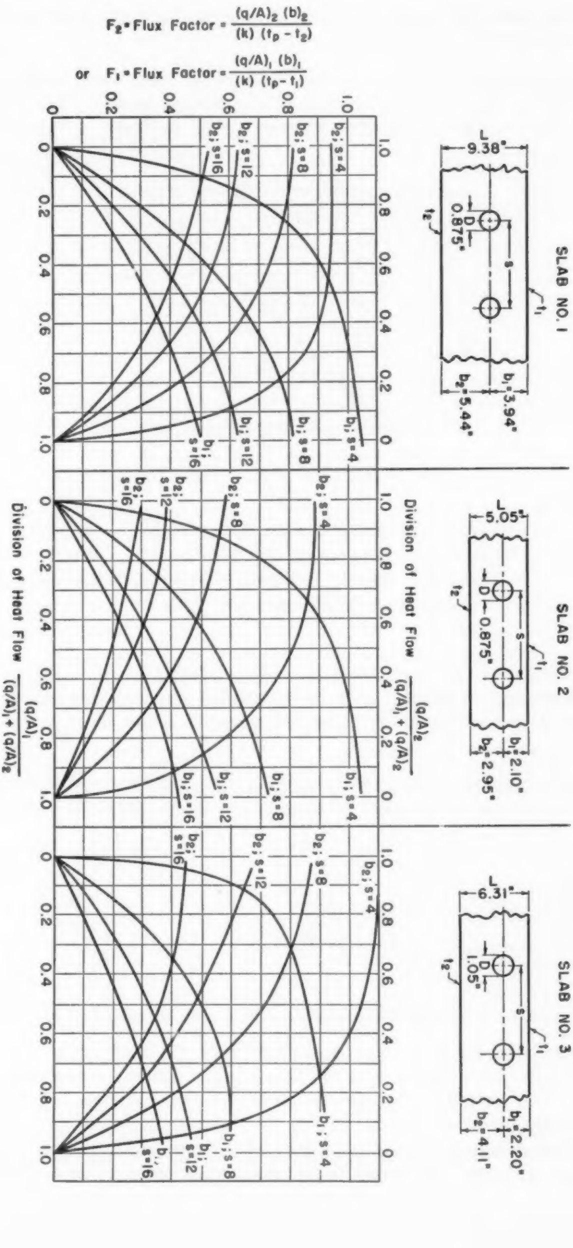

FIG. 5. THERMAL PERFORMANCE OBSERVATIONS FOR SLABS NOS. 1, 2, 3

this case the only heat flow would be from the higher temperature surface to the lower temperature surface.

A surface temperature wave component[3] is a third component entering into the total heat flow rate, but its influence in the present analysis has been neglected. The effect of this component increases with increase in tube spacing

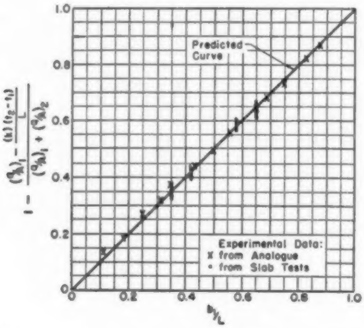

FIG. 6. COMPARISON OF EXPERIMENTAL AND PREDICTED SOURCE HEAT FLOW COMPONENT ON THE BASIS OF SLAB DATA AND ANALOGUE DATA

and decrease in depth of bury. Until the effect of this third component has been investigated and evaluated, it is recommended that applications of these data be limited to the values of D/L and s/L given in Table 2.

TABLE 2—SUGGESTED MAXIMUM VALUES OF D/L AND s/L FOR WHICH THE INFLUENCE OF THE SLAB SURFACE TEMPERATURE WAVE MAY BE NEGLECTED

b/L	D/L	s/L
0.15	0.10	2.0
0.30	0.30	—
0.50	0.80	6.0

Comparison Between Thermal Measurements and Analytical Predictions

The fundamental theory can best be validated by establishing a reasonable agreement between test results and predicted performance within the range covered. This will permit the analytical method to be employed as the basis of practical design material.

The first item of comparison is the rate of heat flow through the slab surfaces. The data give the total mean rates of heat flow through these surfaces.

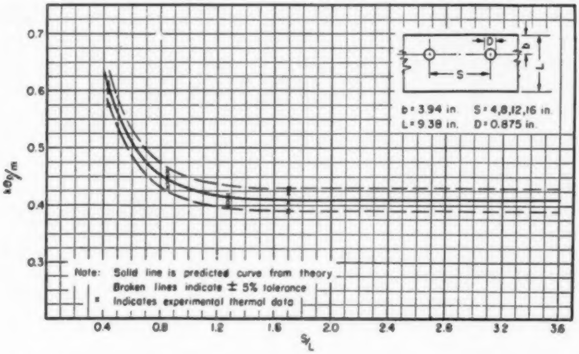

FIG. 7. COMPARISON OF EXPERIMENTAL AND PREDICTED SOURCE COMPONENTS OF THE PIPE-TO-MEAN-SURFACE-TEMPERATURE DIFFERENCE FOR SLAB NO. 1

The linear component is computed and subtracted from the totals to leave an experimentally-determined source component. Fig. 6 shows the comparison of the experimental and predicted source-component quantities; a narrow band along the predicted line would cover all valid test data. Here the theory is judged to be well confirmed within the limits investigated.

The second item of comparison is the difference between the pipe and the mean slab surface temperatures. Here likewise an experimental determination of the source component of this difference is obtained by subtracting a computed linear-flow contribution from the measured difference. Figs. 7, 8 and 9 compare the results with the predictions, where θ_p is the source-component tem-

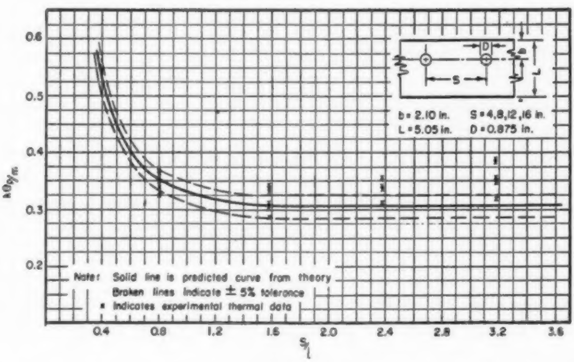

FIG. 8. COMPARISON OF EXPERIMENTAL AND PREDICTED SOURCE COMPONENTS OF THE PIPE-TO-MEAN-SURFACE-TEMPERATURE DIFFERENCE FOR SLAB NO. 2

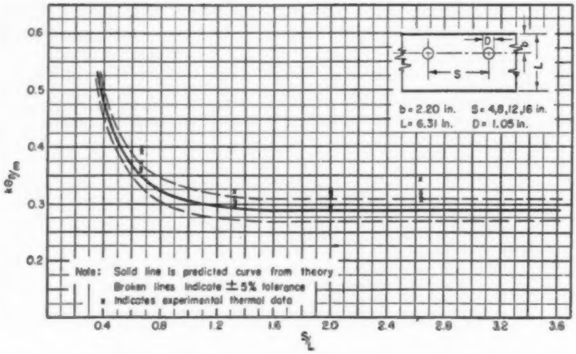

FIG. 9. COMPARISON OF EXPERIMENTAL AND PREDICTED SOURCE COMPONENTS OF THE PIPE-TO-MEAN-SURFACE-TEMPERATURE DIFFERENCE FOR SLAB NO. 3

perature difference, k is the slab thermal conductivity in foot units, and m is the Btu per (hour) (linear foot) heat flow from the pipe. Considering that the experimental magnitudes of θ_p are probably valid only to $\pm$ 1 deg, this places a $\pm$5 percent tolerance on the predicted curve for the magnitudes of k and m involved in the tests; the dashed lines indicate this tolerance.

The rise of the experimental points above the predicted curves at higher magnitudes of s/L, Figs. 8 and 9, is a consequence of having neglected the surface temperature wave component of the slab temperature field in the theory developed thus far[3]. Preliminary estimates of the effect of including this com-

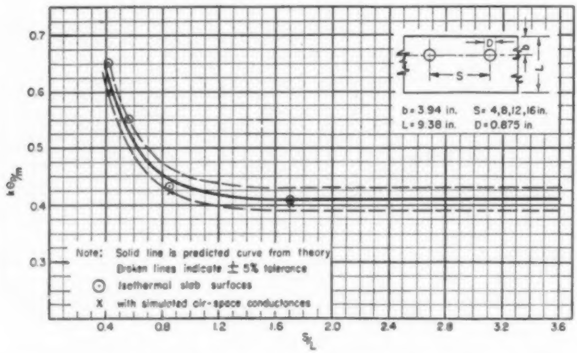

FIG. 10. COMPARISON OF ANALOGUE DATA AND PREDICTIONS OF THE SOURCE COMPONENT OF THE PIPE-TO-MEAN-SURFACE-TEMPERATURE DIFFERENCE FOR SLAB NO. 1

ponent bring the experimental results and the predictions into good agreement over the entire range covered. The consistently high location of the mean trend of the experimental points in Fig. 9 is most probably due to an accidental departure from the supposed depth of bury, or possibly to an erroneous thermal conductivity.

These results are considered to validate the theory, subject to the limitations mentioned.

Comparison Between Analogue Measurements and Analytical Predictions

Once properly adjusted on the basis of thermal data to simulate the slab system, the analogue becomes a convenient tool for extending the range of the comparisons between predictions and experiments.

Again the first comparison is for the source component of the heat flow, Here it is most convenient to operate the analogue with both slab boundaries represented by electrodes maintained at the same voltage, *i.e.* isothermal slab surfaces at the same temperature. Fig. 6 shows this comparison and indicates satisfactory agreement between analogue results and the fundamental theory thereby giving confidence to both the theory and the use of the analogue.

The source component of the pipe-to-surface temperature difference is the subject of the comparison in Fig. 10 wherein slab No. 1 has been simulated. The first series of points shows the measurements obtained with equipotential (isothermal) slab surfaces. Since the theory is exact for this condition, the departure of the points from the predicted curve indicates slight imperfections in the analogue technique.

The second set of points was obtained with an analogue simulation of the air-space conductance prevailing in the corresponding thermal tests. The essential agreement between the two sets of points indicates that in the analogue representation of slab No. 1, as in the thermal tests, the surface voltage (temperature) wave has a very minor influence; hence, data correlation is satisfactorily based upon mean surface conditions.

The laboratory analogue was not an elaborate one, and it is judged that a ± 5 percent tolerance upon its indications is reasonable. This is of the same order as for the thermal tests.

Use of the analogue to demonstrate the theory in a region not covered by the thermal tests is exemplified by Fig. 11. Here the effect of a changing ratio of pipe diameter to slab thickness is indicated for the source component of the pipe-to-surface temperature difference, θ_p. The experimental points were obtained with equipotential (isothermal) slab boundaries.

To summarize, analogue measurements have been shown to agree well with the theory for variations of each of the three geometrical ratios, b/L, s/L, and D/L. These measurements, moreover, were also demonstrated by actual test to be consistent for different values of k, θ_p and m, so long as the modulus $(k\theta_p/m)$ had the same magnitude.

Effect of Insulation Over Slab Surface

So long as the surface-temperature-wave components of the heat flow rate and of the pipe-to-surface temperature difference are negligible (which is the

range within which this report is limited) no modifications of the fundamental analysis or its interpretations are necessary to account for insulation as far as the slab performance, in itself, is concerned. The slab surface conditions are still adequately represented by a mean surface temperature, and the addition of insulation merely interposes a certain thermal resistance between the slab surface and the environment. In the steady state all of the heat passing through the slab surface also passes through the insulation surface, so that the difference

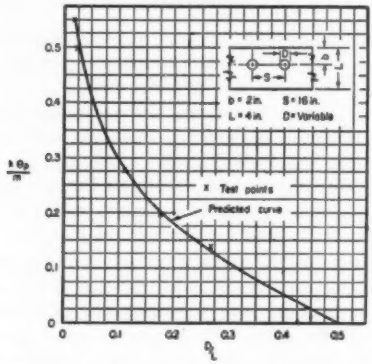

FIG. 11. COMPARISON OF ANALOGUE DATA AND PREDICTIONS OF THE SOURCE COMPONENT OF THE PIPE-TO-MEAN-SURFACE-TEMPERATURE DIFFERENCE FOR VARYING RATIOS OF D/L

between the mean slab surface temperature t_1 and the mean insulation surface temperature t_i is

$$(t_1 - t_i) = (q/A)_1 R_i$$

where

R_i = the insulation thermal resistance, (Fahrenheit degrees) (square foot) (hour) per (Btu).

Adding surface insulation with a given environment will change the mean temperature and rate of heat flow through the side of the slab originally exposed to this environment; and, other conditions remaining unchanged, this will produce a corresponding alteration on the far side of the slab. This conclusion has been validated by both thermal and analogue test data.

CONCLUSIONS

Experimental studies of heat flow in concrete slabs, and data obtained by means of an electrical analogue, have confirmed the fundamental theory developed for the analysis of heat flow in panel heating sections[3].

The thermal test system designed and operated at the Research Laboratory has proved capable of yielding trustworthy data.

The electrical analogue has been shown to be a convenient means for extending the range of data obtained by thermal tests.

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1. Performance of Heating Panel Studied (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping & Air Conditioning*, April 1949, p. 112).

2. A.S.H.V.E. RESEARCH REPORT No. 1389—Electric Analogger Studies on Panels with Imbedded Tubes, by Carl F. Kayan (A.S.H.V.E. TRANSACTIONS, Vol. 56, 1950, p. 205).

3. A.S.H.V.E. RESEARCH REPORT No. 1388—Heat Flow Analysis in Panel Heating or Cooling Sections, Case I—Uniformly Spaced Pipes Buried Within a Solid Slab, by L. E. Hulbert, H. B. Nottage, and C. V. Franks (A.S.H.V.E. TRANSACTIONS. Vol. 56, 1950, p. 189).

DISCUSSION

P. B. GORDON, New York: This first presentation by the Laboratory has to do with what we call *Group A* activities, the study of the performance of heat flow characteristics within the panel. Of course, in the overall program, we must have the information from the panel surface to the room with all the other problems involved; but it was felt several years ago that unless we split the problem up into smaller segments, we would lose control of the overall study.

As such, this only forms a part of the *Group A* work. There is other work being done in plaster panels, and on heat flow from the underside of the concrete slab into the earth. When these and other studies are completed, we have the problem of integrating and correlating these data, and eventually preparing design and application procedures and techniques. The Laboratory is to be complimented on the amount of work completed to date.

W. A. DANIELSON, Memphis, Tenn.: Some 25 years ago tests were made on highways in Illinois called the Baked Roads Tests. These showed some very remarkable arching effects. I wonder if you measured any distortion in the concrete itself when you made the tests. That might become serious when dealing with structural problems.

W. S. HOBBS, Philadelphia, Pa.: I have been working with heating panels since 1941. Has the effect of surface reinforcing steel been included in the test data?

D. L. MILLS, Rome, N. Y.: For the benefit of those not familiar with panel heating, I suggest that Mr. Humphreys explain why there are idle tubes in the slab, because in actual practice, in a panel heating concrete floor slab there would be no idle tubes.

AUTHORS' CLOSURE (C. M. Humphreys): General Danielson has raised a point which is of considerable interest to the structural engineer. However, our work was confined to studies of heat transfer, and no attempt was made to measure stresses in the slab.

In answer to Mr. Hobbs' question, there was no reinforcing steel in the slabs tested. It was felt that this would only add another variable.

As Mr. Mills has suggested, the ordinary slab does not contain idle tubes. However, one of the variables which we wanted to investigate was the effect of tube spacing.

Since the construction of six foot square slabs is expensive, both in money and time, test equipment and slabs were so designed that each slab would yield the maximum amount of test data. The absorbers were made 48 in. square, and since this figure is divisible by 4, 8, 12, and 16, it was possible by valving off a part of the tubes, to make tests with each slab at each of these tube spacings. The tubes not in use during tests at the wider tube spacings were *idle* tubes. Professor Kayan has demonstrated by analogue studies, that these idle tubes had no significant effect on the heat transfer within the slabs.

1388

No. 1388

HEAT FLOW ANALYSIS IN PANEL HEATING OR COOLING SECTIONS†

Case I—Uniformly Spaced Pipes Buried Within a Solid Slab

By L. E. Hulbert*, H. B. Nottage** and C. V. Franks††, Cleveland, Ohio

This paper is the result of research carried on by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at its Research Laboratory located at 7218 Euclid Ave., Cleveland 3, Ohio.

The objective of the A.S.H.V.E. program of panel heating and cooling research is the development of sound and acceptable design data and design methods which may be used in the majority of present day applications[1]. The program is broad in its conception and, for purposes of convenience, was so organized that it could be broken down into a number of important constituent parts which could be investigated independently and concurrently. One of these main divisions includes the study of heat flow within a panel.

This paper is the first of a group dealing with the *theory* of heat flow in a slab. The development of analytical solutions of basic problems of heat flow will save experimental time and permit application of the data beyond the limits covered entirely by experiment. A second paper dealing with heat flow analysis for floor slabs on earth will be prepared.

The correlation between the theory developed in this paper and the results obtained experimentally on concrete slabs tests at the Society's Research Laboratory is discussed in another paper[2].

The results of cooperative research at Columbia University employing its Electric Analogger are reported by Kayan[3]. The part of the program covered by these three papers is being carried on under the general guidance of Group A[△] of the Technical Advisory Committee on Panel Heating and Cooling.

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† Abstract of this report presented at the 56th Annual Meeting of The American Society of Heating and Ventilating Engineers, Dallas, Tex., January 1950.

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1 Exponent numerals refer to References.

△ Personnel of *Group A*—TAC on Panel Heating and Cooling: A. B. Algren, *Chairman;* C. E. Abbey, R. A. Biggs, R. S. Dill, H. L. Flodin, F. E. Giesecke, W. S. Harris, C. F. Kayan, G. D. Lain, R. L. Maher, C. W. Meininger, D. L. Mills, C. W. Nessell, P. S. Park, S. I. Rottmayer, E. E. Scott, S. K. Smith, R. K. Thulman, S. M. Van Kirk, W. J. Widmer, G. L. Wiggs.

SYSTEM ANALYZED

Fig. 1 shows the slab system as set up for analysis. As a basis for reasoning, steady-state conditions are assumed in a two-dimensional region of constant thermal conductivity and uninfluenced by possible edge effects. The mean surface temperature is defined as the mean along a line running perpendicular to the pipe axis.

Use of a line source to represent a pipe is permissible in the range where isotherms about the source are essentially circles corresponding to the tube outer diameter. Since departures occur when the tube is tangent to or very close to the slab surface, it is planned to consider this effect and develop a

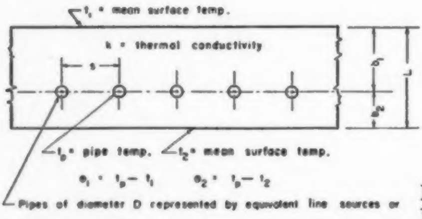


FIG. 1. THE SLAB SYSTEM ANALYZED

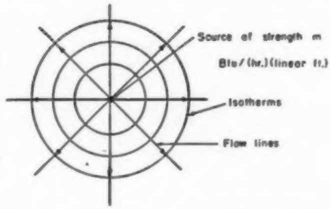


FIG. 2. FLOW-LINE AND ISOTHERM NETWORK FOR A LINE SOURCE OF STRENGTH m IN AN INFINITE MEDIUM

method of calculation to compensate for it in a subsequent paper. For the present the analysis given here should *not* be applied for ratios of pipe outer diameter D, to slab thickness L, greater than those given in Table 1, where b denotes the distance between the pipe centerline and the slab surface concerned. If these limits are exceeded, the predicted difference between the maximum and minimum temperature about the tube circumference becomes greater than 2.5 deg.

Another paper[2] offers some interim suggested limits on s/L, the ratio of pipe spacing to slab thickness, which may be followed in applying the theory of this paper.

FUNDAMENTAL CONCEPTS OF METHOD OF ANALYSIS

The concepts basic to the theory are treated extensively by Hague[4].

A first concept is the use of sources and images or sinks in analyzing heat flow. Consider an infinite medium of thermal conductivity k containing a line

TABLE 1—SUGGESTED MAXIMUM LIMITS OF D/L FOR USE OF THE THEORY OF THIS PAPER

b/L	*Max.* D/L
0.15	0.10
0.30	0.30
0.50	0.80

source of strength m, *i.e.*, emitting m Btu per (hr) (linear ft). Fig. 2 shows a section taken perpendicular to the line source. The isotherms here will be circles about the source, and the temperature difference between two isotherms of radii r_1 and r_2, where $r_2 > r_1$, will be

$$\theta = t_1 - t_2 = -\frac{m}{4\pi k} \log_e \left(\frac{r_1}{r_2}\right)^2 \tag{1}$$

Next suppose that at some one point in the medium the temperature is known, and call this the *datum temperature*, t_d. A coordinate system may be located in the medium by taking the origin at the point of known temperature and passing the y-axis through this point and the source point. The source is then

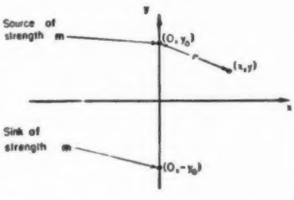


FIG. 3. COMBINATION OF A LINE SOURCE AND LINE SINK WHICH YIELDS THE TEMPERATURE FIELD IN A SEMI-INFINITE MEDIUM HAVING AN ISOTHERMAL SURFACE ALONG THE X-AXIS

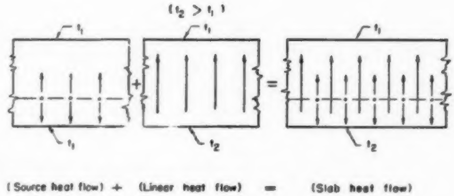


FIG. 4. PRINCIPLE OF SUPERPOSITION

at the point $(0, y_o)$, Fig. 3, and r of Equation 1 becomes $\sqrt{x^2 + (y - y_o)^2}$. This establishes a coordinate reference for the temperature field.

If a second source were placed in the medium, its temperature contribution at any point could be referred to the same datum temperature, t_d, at the coordinate origin. Then, by the superposition principle the temperature at any point would be the algebraic sum of the contributions from the two sources, both contributions being referred to t_d. Algebraically, if $(t_a - t_d)$ were the temperature contribution of the first source at some point and $(t_b - t_d)$ that of the second source at the same point, the combination would produce a temperature t given by

$$\theta = (t - t_d) = (t_a - t_d) + (t_b - t_d).$$

Let the superposition be applied to the case of a sink (negative source) of strength m being located at the point $(0, -y_o)$; this sink may be regarded as the *image* of the source in the coordinate axis. The temperature at any point (x, y) resulting from the source and sink combination is given by

$$\theta = t - t_d = -\frac{m}{4\pi k} \log_e \left(\frac{x^2 + (y - y_o)^2}{x^2 + (y + y_o)^2}\right) \tag{2}$$

In Equation 2, $\theta = 0$ when $y = 0$, which means that the x-axis is an *isotherm* at the datum temperature. Thus, the system of Fig. 3 serves to represent the case of a line source in a semi-infinite conducting medium and located parallel to an isothermal plane boundary. Equation 2 gives the temperature field, restricting y to lie within the medium. Knowing the temperature field and the thermal conductivity, the heat flow rate at any point may be derived mathematically.

Superposing the combination of a second isothermal surface at the same datum temperature and a series of uniformly spaced parallel sources, there results the slab system of Fig. 1, except that the case desired is a slab having *different* mean surface temperatures. The superposition concept can be extended to meet this requirement. Upon the system of the row of sources (pipes), between two isothermal surfaces having the same temperature, there is superposed

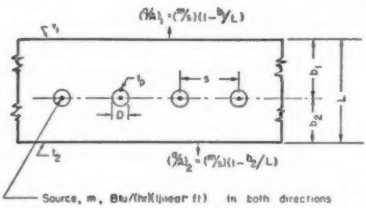


FIG. 5. SCHEMATIC ILLUSTRATION OF HEAT FLOW QUANTITIES

a linear heat flow across the slab as if the sources were not present and the two surfaces were at different temperatures. The heat flow or temperature field in a slab is therefore the resultant of two components which may be defined as follows:

1. *Source component* is the magnitude which would prevail if the slab surfaces were isothermal at the same temperature and the pipes were the only sources of heat.

2. *Linear component* is the magnitude which would prevail if the sources (pipes) were removed and the slab surfaces were isothermal but at different temperatures. In this case the only heat flow would be from the higher temperature surface to the lower temperature surface.

The final combination is represented schematically in Fig. 4 and must be kept clearly in mind as the basis of the present analysis.

ANALYTICAL RESULTS

In this paper results are presented only in a generalized form; the presentation of simplified material for use in practical applications will be made elsewhere. Here both the temperature difference and heat-flow rates are expressed as the sum of two parts, a *source* component and a *linear-flow* component, in accord with the superposition discussed above.

Pipe-to-Surface Temperature Differences

Referring to Fig. 5, the difference between the pipe temperature, t_p, and the mean surface temperature t_1 is

$$(t_p - t_1) = \theta_p + (t_2 - t_1)\left(\frac{b_1}{L}\right) \tag{3}$$

where the subscripts 1 and 2 may be interchanged if desired.

Here the term θ_p is the source contribution and is obtained from the curves of Figs. 6a through 6f once the slab geometry, source strength and slab thermal conductivity are known. (The factor b in Fig. 6 would be taken as b_1 consistent with Fig. 5 and Equation 3). The source strength m in Fig. 6 must always be the *total* heat flow per linear feet of pipe to *both sides* of the slab. The relationship for proportioning this total between the two sides is given in the next section. In an application problem the rates of heat flow through the surface would be further linked to the remainder of the thermal circuit outside of the slab. Arithmetic interpolation will be satisfactory between the curve families of Figs. 6a through 6f. For values of s/L greater than 1.6, the curve for $(s/L) = 1.6$ is to be used so long as the slab surfaces remain isothermal. The term $(t_2 - t_1)$ (b_1/L) in Equation 3 represents the influence of the linear-flow component referred to the center of the tube.

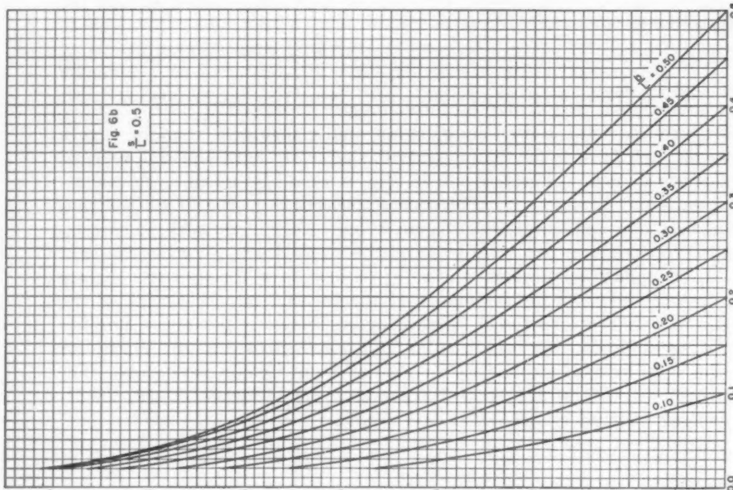
NOMENCLATURE

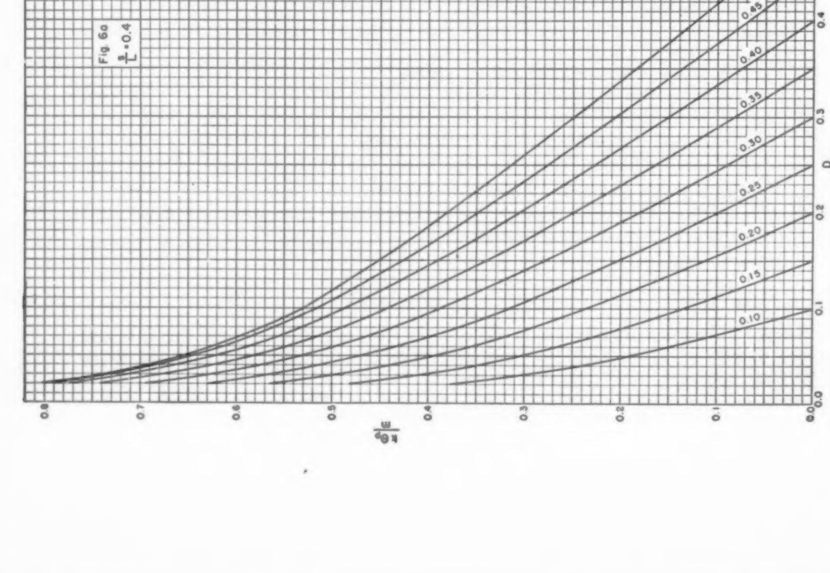
- b = distance from pipe centerline to slab surface, inches, designated as b_1 and b_2 for the two slab surfaces, respectively.
- D = outer pipe diameter, inches.
- k = slab thermal conductivity, Btu per (hour) (square foot) (Fahrenheit degree per foot).
- L = slab thickness, inches.
- m = pipe heat emission (or reception) rate, or source (or sink) strength, Btu per (hour) (linear foot).
- $(q/A)_1$ = mean heat flow rate through slab surface having temperature t_1, Btu per (hour) (square foot of surface).
- $(q/A)_2$ = mean heat flow rate through slab surface having temperature t_2, Btu per (hour) (square foot of surface).
- r_1, r_2 = radial distances from source, inches.
- s = centerline pipe spacing, inches.
- t_1, t_2 = mean slab surface temperatures, Fahrenheit degrees.
- t_d = datum temperature, Fahrenheit degrees.
- t_p = mean pipe outer surface temperature, Fahrenheit degrees.
- θ = $(t - t_d)$, in general, Fahrenheit degrees.
- θ_p = $(t_p - t_1)$ or $(t_p - t_2)$, as desired, Fahrenheit degrees.
- x, y = coordinate dimensions, inches.
- y_0 = source coordinate, Fig. 3, inches.

Heat Flow Rates

From the mathematically exact derivation it is established that the division of the *source component* of the heat flow between the two slab surfaces, with reference to Fig. 5, is:

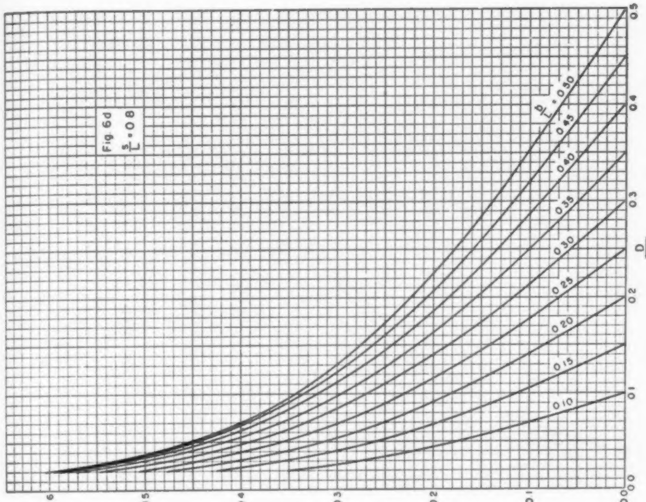
$m\ (1 - b_1/L)$ To the surface at t_1.
$m\ (1 - b_2/L)$ To the surface at t_2.

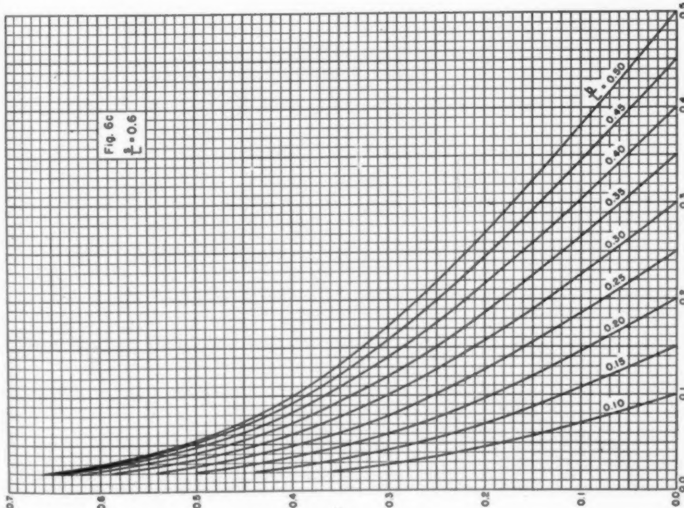




FIGS. 6a AND 6b. RELATIONSHIP BETWEEN THERMAL VARIABLES AND SLAB GEOMETRY

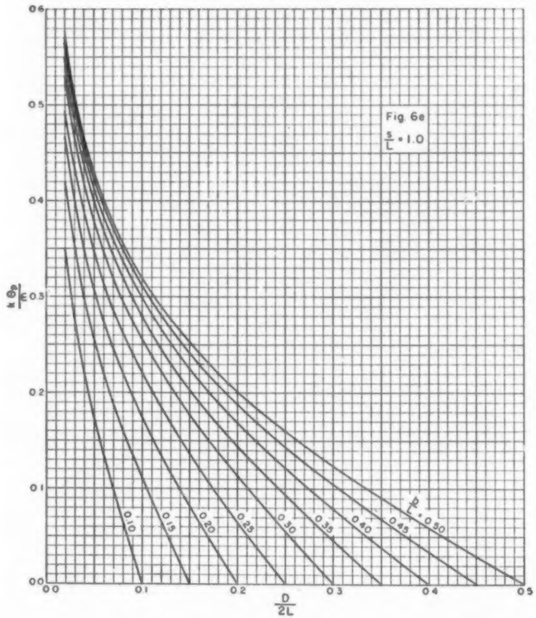
$\left(\frac{k\theta_p}{m}\right), \left(\frac{s}{L}\right), \left(\frac{b}{L}\right), \left(\frac{D}{2L}\right)$ (for values of $s/L = 0.4$ to 0.8)

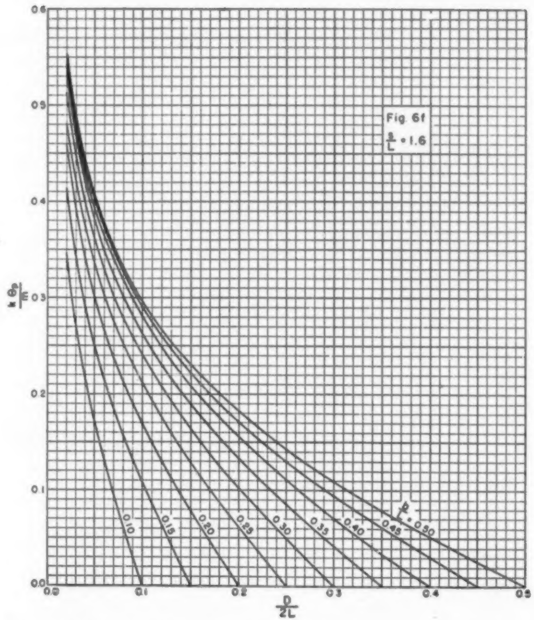




FIGS. 6c AND 6d. RELATIONSHIP BETWEEN THERMAL VARIABLES AND SLAB GEOMETRY

$\left(\frac{k\theta p}{m}\right), \left(\frac{s}{L}\right), \left(\frac{b}{L}\right), \left(\frac{D}{2L}\right)$ (for values of s/L = 0.6 to 0.8





FIGS. 6e AND 6f. RELATIONSHIP BETWEEN THERMAL VARIABLES AND SLAB GEOMETRY

$\left(\frac{k\theta_D}{m}\right)$, $\left(\frac{s}{L}\right)$, $\left(\frac{b}{L}\right)$, $\left(\frac{D}{2L}\right)$ (for values of $s/L = 1.0$ and 1.6)

The heat flow rate through either one of the slab surfaces is computed as the sum of a source component and a linear-flow component. Supposing that $t_2 > t_1$, this may be expressed as follows:

$$(q/A)_1 = \left[(q/A)_1 + (q/A)_2\right]\left[1 - \frac{b_1}{L}\right] + \frac{k\,(t_2 - t_1)}{L}, \text{ Btu/(hr) (sq ft surface)} \quad . . (4)$$

The source strength, Btu/per (hour) (linear foot pipe), is related to the total heat flow by the expression

$$\frac{m}{s} = (q/A)_1 + (q/A)_2 \qquad (5)$$

where

s = the pipe centerline spacing.

Expressing Equation 4 per linear foot of pipe,

$$(q/l)_1 = m\left(1 - \frac{b_1}{L}\right) + ks\left(\frac{t_2 - t_1}{L}\right), \text{ Btu/(hr) (linear ft pipe)} \qquad (6)$$

The subscripts 1 and 2 may be interchanged if desired in Equations 4 and 6.

It must be emphasized again that in application the mean surface temperatures t_1 and t_2 will find their levels according to the remainder of the thermal circuit outside of the slab.

Although, pending further analytical studies, the suggested limits of Table 1 must remain, calculations of *heat flow* for the case where the pipe is very close to the surface may be made with greater confidence than calculations of *(mean surface) temperature*. The reason for this is that, for most heating applications the heat-flow rate from the surface may be taken as directly proportional to the difference between the surface temperature and an environmental temperature; and with this direct proportionality, surface temperature waves may be replaced by the corresponding mean temperature in computing heat flow on the air side of the surface.

EXPERIMENTAL CONFIRMATION

As previously indicated, the results of the experimental investigations are summarized in another paper[2]. In this presentation confirmation of the theory given here will be limited to Equation 4.

From the test data obtained in the experimental studies[2] mean surface temperatures and rates of heat flow were determined. By rearranging Equation 4 a check on the theory can be made by plotting

$$\left[1 - \frac{(q/A)_1 - k\,\dfrac{t_2 - t_1}{L}}{(q/A)_1 + (q/A)_2}\right] vs.\ (b_1/L)$$

The comparison of theoretical and experimental data is shown in Fig. 7 from which it may be seen that experimental scattering from the theory is relatively small and of the magnitude to be expected in the type of work involved. Hence, the theory is considered to have been confirmed within the limits indicated.

It is of practical importance to note that the heat flow is quite sensitive to

the ratio (b_1/L). This means that pipe bury should be held to careful tolerance in construction work if design calculations are to apply accurately.

EXAMPLE

A numerical example, duplicating one of the conditions for which test data were taken, will illustrate the use of Equation 3 for calculating pipe temperature. Geometrical factors are: $s = 8$ in., $L = 5.05$ in., $(b_1/L) = 0.416$, $s/L = 1.58$, $D/2L = 0.0865$. Slab thermal conductivity, $k = 1.09$ Btu per (hr) (sq ft) (F deg) (ft). Measured heat flow rates, $(q/A)_1 = 48.0$ and $(q/A)_2 = 3.28$ Btu per

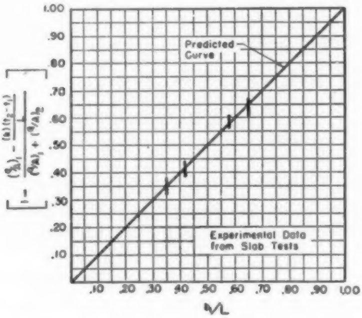


FIG. 7. COMPARISON OF THEORY AND EXPERIMENT

(hr) (sq ft). Measured mean surface temperatures, $t_1 = 106.9$ and $t_2 = 114.5$ F. Compute m from Equation 5,

$$m = (48.0 + 3.28)\left(\frac{8}{12}\right) = 34.2 \text{ Btu per (hr) (linear ft)}$$

From Fig. 6,

$$\frac{k\,\theta_p}{m} = 0.306, \text{ or } \theta_p = \frac{0.306 \times 34.2}{1.09} = 9.60 \text{ F}$$

Compute the pipe temperature from Equation 3,

$$\begin{aligned} t_p &= t_1 + \theta_p + (t_2 - t_1)(b_1/L). \\ &= 106.9 + 9.60 + (7.6)(0.416). \\ &= 106.9 + 9.60 + 3.16 = 119.7 \text{ F}. \end{aligned}$$

t_p by experiment is 120.6 F which is satisfactory agreement. Numerous other check calculations for test conditions also have yielded agreements in the order of 1 deg. Since the calculations are quite sensitive to uncertainties in (b_1/L) a small dimensional error has a large effect upon the computed temperatures.

CONCLUSIONS

1. Using the source-image principle, a workable analysis has been developed for heat flow in the system of a row of pipes buried within a slab.

2. The analytical predictions have been confirmed by test results as is demonstrated further in another paper[2].

3. The theory can be used for design calculations until such time as more complete practical design data and design methods are available. However, if it is to be used for this purpose, it is essential that the user have a thorough fundamental understanding of the theory.

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APPENDIX

TEMPERATURE FIELD FOR A SLAB WITH ISOTHERMAL SURFACES AT DIFFERENT TEMPERATURES AND CONTAINING A UNIFORMLY-SPACED ROW OF PIPES SERVING AS HEAT SOURCES OR SINKS

Referring to Fig. 3 and Equation 2 of the paper, the temperature field is given for a line source in a semi-infinite medium located parallel to an isothermal plane surface which is at a datum (which may be taken as zero) temperature. In this portrayal the source is combined with a sink of the same strength located at the image of the source in the boundary.

Following the superposition principle further, consider next a source of strength m within a slab of which both surfaces are isothermal at the datum temperature, Fig. 8. As before, the theory is developed to make, say, the lower boundary isothermal at the datum temperature by first placing a sink of strength m at the image point of the source in the lower surface and considering the slab to be part of an infinite medium. However, the sink so placed will affect the temperature of the upper boundary; and to nullify this, another source of strength m is needed at the image point of the sink in the upper boundary. Again, this source affects the temperature of the lower boundary, so that it becomes necessary to repeat the process *ad infinitum.*

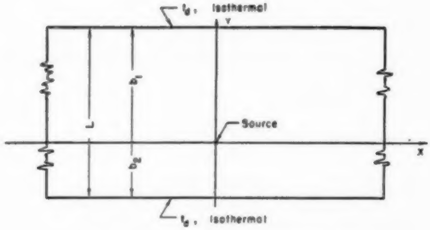

FIG. 8. LINE SOURCE IN A SLAB

TABLE 2—POSITIONS AND SIGNS OF SOURCES AND SINKS ALONG y-AXIS

	Reflections in $y = b$, (Fig. 8)		Reflections in $y = (b - L)$, (Fig. 8)	
	y	*Sign*	y	*Sign*
Original source	0	+	0	+
1st reflection	$2b$	−	$2b - 2L$	−
2nd reflection	$-2L$	+	$2L$	+
3rd reflection	$2b + 2L$	−	$2b - 4L$	−
4th reflection	$-4L$	+	$4L$	+
5th reflection	$2b + 4L$	−	$2b - 6L$	−

In a similar manner to make the upper boundary isothermal at the datum temperature a sink of strength m is placed at the image point of the source in the upper boundary. This sink affects the temperature of the lower boundary, and so on.

The temperature field in the slab is finally represented by a source at the coordinate origin and an infinite string of both sources and sinks lying in both directions along the y-axis. The positions and signs of the first few are given in Table 2; a (+) sign represents a source, and a (−) sign a sink.

The temperature at any point (x, y) in the slab will be the sum of all the contributions from these sources and sinks.

For a single source,

$$\theta = t - t_d = -\frac{m}{4\pi k}\log_e r^2 \tag{7}$$

Summing for all,

$$\begin{aligned}\frac{\theta\, 4\pi k}{m} = &-\log_e(x^2 + y^2) + \log_e[x^2 + (y - 2b)^2] + \log_e[x^2 + (y - 2b + 2L)^2] \\ &- \log_e[x^2 + (y + 2L)^2] - \log_e[x^2 + (y - 2L)^2] \\ &+ \log_e[x^2 + (y - 2b - 2L)^2] + \ldots\end{aligned} \tag{8}$$

Grouping positive and negative terms together,

$$\begin{aligned}\frac{\theta\, 4\pi k}{m} = &\log_e[x^2 + (y - 2b)^2][x^2 + (y - 2b + 2L)^2][x^2 + (y - 2b - 2L)^2] \\ &- \log_e[x^2 + y^2][x^2 + (y - 2L)^2][x^2 + (y + 2L)^2][x^2 + (y - 4L)^2]\,.\end{aligned} \tag{9}$$

To simplify, consider the products only. In the first product let:

$$\begin{aligned}(y - 2b) &= Z \\ 2L &= a,\end{aligned}$$

and the relation becomes

$$[x^2 + Z^2][x^2 + (Z + a)^2][x^2 + (Z - a)^2][x^2 + (Z - 2a)^2] \;\ldots \tag{10}$$

Similarly in the second product let:

$$\begin{aligned}y &= Z_1 \\ 2L &= a,\end{aligned}$$

and the relation becomes

$$[x^2 + Z_1^2][x^2 + (Z_1 - a)^2][x^2 + (Z_1 + a)^2][x^2 + (Z_1 - 2a)^2] \;\ldots \tag{11}$$

Equations 10 and 11 are of the same form and may be simplified by manipulation.

Consider Equation 10,

$$[x^2 + Z^2]\,[x^2 + (Z-a)^2]\,[x^2 + (Z+a)^2] \;.\;.\;.\;.$$
$$= \left[1 + \frac{x^2}{Z^2}\right]\left[1 + \frac{x^2}{(Z-a)^2}\right]\left[1 + \frac{x^2}{(Z+a)^2}\right] \;.\;.\; (Z^2)\,(Z-a)^2\,(Z+a)^2 \qquad (12)$$

The first product in Equation 12 is known** to be

$$\frac{\cosh \dfrac{2\pi x}{a} - \cos \dfrac{2\pi Z}{a}}{2 \sin^2 \dfrac{\pi Z}{a}} \;.\;.\;.\;. \qquad (13)$$

The second product in Equation 12 is.

$$Z^2\,(Z-a)^2\,(Z+a)^2\,(Z-2a)^2\,(Z+2a)^2 \;.\;.\;.\;.$$
$$= Z^2\,(a^2 - Z^2)^2\,(2^2\,a^2 - Z^2)^2\,(3^2\,a^2 - Z^2)^2 \;.\;.\;.\;.$$
$$= \left\{\frac{Z^2}{a^2}\left(1 - \frac{Z^2}{a^2}\right)^2\left(1 - \frac{Z^2}{2^2\,a^2}\right)^2\left(1 - \frac{Z^2}{3^2\,a^2}\right)^2 \;.\;.\;.\;.\right\}\; a^2\,a^4\,2^4\,a^4\,3^4\,a^4 \;,\;.\;.\;.,$$

which is recognized to be

$$\frac{1}{\pi^2}\left(\sin^2 \frac{\pi Z}{a}\right) a^2\,a^4\,2^4\,a^4\,3^4\,a^4 \;.\;.\;.\;. \qquad (14)$$

Combining Equations 13 and 14 gives for the product Equation 10,

$$\frac{\cosh \dfrac{2\pi x}{a} - \cos \dfrac{2\pi Z}{a}}{2 \sin^2 \dfrac{\pi Z}{a}}\left(\frac{1}{\pi^2} \sin^2 \frac{\pi Z}{a}\right) a^2\,a^4\,2^4\,a^4\,3^4\,a^4 \;.\;.\;. \qquad (15)$$

These results allow Equation 9 for $\dfrac{\theta 4\pi k}{m}$ to be evaluated:

$$\frac{\theta 4\pi k}{m} = \log_e\left[\cosh \frac{\pi x}{L} - \cos \frac{\pi\,(y-2b)}{L}\right] - \log_e\left[\cosh \frac{\pi x}{L} - \cos \frac{\pi y}{L}\right]$$
$$+ \log_e\left[\frac{1}{2\pi^2}\,(2L)^2\,(2L)^4\,(2)^4\,(2L)^4\,(3)^4 \;.\;.\;.\;.\right]$$
$$- \log_e\left[\frac{1}{2\pi^2}\,(2L)^2\,(2L)^4\,(2)^4\,(2L)^4\,(3)^4 \;.\;.\;.\;.\right]$$

Finally,

$$\theta = \frac{m}{4\pi k}\left\{\log_e\left[\cosh \frac{\pi x}{L} - \cos \frac{\pi\,(y-2b)}{L}\right] - \log_e\left[\cosh \frac{\pi x}{L} - \cos \frac{\pi y}{L}\right]\right\} \qquad (16)$$

This completes the temperature field for a single source in the slab. The desired solution is that for a uniformly spaced row of sources along the x-axis, Fig. 1. Considering the nth source with center at point $(x_n, 0)$, its temperature field is given by

$$\theta_{n(x,y)} = \frac{m}{4\pi k}\left\{\log_e\left[\cosh \frac{\pi(x-x_n)}{L} - \cos \frac{\pi(y-2b)}{L}\right]\right.$$
$$\left. - \log_e\left[\cosh \frac{\pi(x-x_n)}{L} - \cos \frac{\pi y}{L}\right]\right\} \qquad (17)$$

** A Treatise on Plane Trigonometry, by E. W. Hobson *(Cambridge University Press)*, Fourth Edition, 1918, p. 353.

The temperature at point (x, y) produced by all tubes will be the summation,

$$\theta_{(x,y)} = \sum_{n=-\infty}^{+\infty} \theta_{n(x,y)} \qquad (18)$$

The curves for θ_p, Fig. 6 of the paper, were calculated by summing overall sources which had a noticeable effect; fortunately, the series converges rapidly. The coordinates (x, y) at which θ_p was determined are $(x = 0,\ y = D/2)$, which places the point on the outer circumference of the pipe of diameter D and on the diameter running transverse to the slab at the point nearest to the slab surface concerned.

Referring further to Equation 3 of the paper, the superposed linear temperature component has been located at the center of the pipe. This is considered the best choice because the centerpoint would average the linear variation across the pipe diameter.

The calculations which were made to check how closely circular were the isotherms about the line source at the distance of the pipe radius involved a comparison between θ computed for $(x = 0,\ y = D/2)$ and for $(x = 0,\ y = -D/2)$. The suggested limits of Table 1 in the paper were established thereby. Further modifications of the theory to cover conditions beyond the tentative limits of Table 1 will be presented in a subsequent paper.

AREA-MEAN HEAT FLOW RATES OVER THE SLAB SURFACES

Consider only the source component of heat flow and take a plane section parallel to the slab surface, at $y = y_o$ and of unit length along the source lines, across which the heat flow rate is desired.

The heat flow rate per unit area in any direction at a point is equal to the product of the thermal conductivity and the temperature gradient in the desired direction. Taking the temperature field given by Equations 17 and 18, the heat flow rate per unit area in the y-direction at any point is

$$\left(\frac{q}{A}\right)_y = -k\frac{\partial \theta_{(x,y)}}{\partial y} = -\frac{m}{4\pi}\sum_{n=-\infty}^{+\infty}\left[\frac{\frac{\pi}{L}\sin\frac{(y-2b)\pi}{L}}{\cosh\frac{(x-x_n)\pi}{L} - \cos\frac{(y-2b)\pi}{L}} - \frac{\frac{\pi}{L}\sin\frac{y\pi}{L}}{\cosh\frac{(x-x_n)\pi}{L} - \cos\frac{y\pi}{L}}\right] \quad . \ . \ . \ (19)$$

To find the total flow rate through a plane segment at $y = y_o$ and over the interval $w \leqq x \leqq w_1$, y is held constant while Equation 19 is integrated with respect to x from $x = w$ *to* $x = w_1$,

$$q_{\left(\substack{w\to w_1\\ y=\text{constant}}\right)} = -\frac{m}{4L}\int_{w}^{w_1}\sum_{n=-\infty}^{+\infty}\left[\frac{\sin\frac{(y-2b)\pi}{L}}{\cosh\frac{(x-x_n)\pi}{L} - \cos\frac{(y-2b)\pi}{L}} - \frac{\sin\frac{y\pi}{L}}{\cosh\frac{(x-x_n)\pi}{L} - \cos\frac{y\pi}{L}}\right]dx. \quad . \ . \ (20)$$

The variable x occurs only in the denominator of the two fractions. Each fraction is integrable with respect to x, y remaining constant. The result is

$$q_{\left(\substack{w\to w_1\\ y=\text{const.}}\right)} = -\frac{m}{2\pi}\sum_{n=-\infty}^{+\infty}\left[\tan^{-1}\left(\frac{e^{\frac{(x-x_n)\pi}{L}} - \cos\frac{(y-2b)\pi}{L}}{\sin\frac{(y-2b)\pi}{L}}\right) - \tan^{-1}\left(\frac{e^{\frac{(x-x_n)\pi}{L}} - \cos\frac{y\pi}{L}}{\sin\frac{y\pi}{L}}\right)\right]_{w}^{w_1} \quad . \ . \ (21)$$

Because of symmetry, the total heat flow rate will be the same for each x- interval between a source and at the mid-point of the source spacing distance. The area-mean flow rate over the *slab surface* is of primary interest; this is obtained by dividing

the total flow by the area involved in half a spacing, $s/2$ (1) sq ft. For this determination the limits on Equation 21 will be taken as $w = 0$, $w = s/2$. At the surface. $y = b$.

$$(q/A)_{mean} = -\frac{m}{\pi s} \sum_{n=-\infty}^{+\infty} \left[\tan^{-1}\left(\frac{e^{-\frac{x_n \pi}{L}} - \cos\frac{b\pi}{L}}{\sin\left(-\frac{b\pi}{L}\right)} \right) - \tan^{-1}\left(\frac{e^{-\frac{x_n \pi}{L}} - \cos\frac{b\pi}{L}}{\sin\frac{b\pi}{L}} \right) \right.$$
$$\left. - \tan^{-1}\left(\frac{e^{\frac{(s/2 - x_n)\pi}{L}} - \cos\frac{b\pi}{L}}{\sin\left(\frac{b\pi}{L}\right)} \right) + \tan^{-1}\left(\frac{e^{\frac{(s/2 - x_n)\pi}{L}} - \cos\frac{b\pi}{L}}{\sin\frac{b\pi}{L}} \right) \right]$$

To achieve a further combination of the terms, recall that the origin ($x = 0$, $y = 0$) was placed at one of the source points. Successive values of x_n for a source spacing s are:

n	x_n
0	0
x_1	s
x_2	$2s$
$x(-_1)$	$-s$
$x(-_2)$	$-2s$
x_n	ns

where n takes on all integral values from $-\infty$ to $+\infty$. The source component of the surface heat flow rate is then

$$(q/A)_{mean} = \frac{2m}{\pi s} \sum_{n=-\infty}^{+\infty} \left[\tan^{-1}\left(\frac{e^{\frac{(s/2 - ns)\pi}{L}} - \cos\frac{b\pi}{L}}{\sin\frac{b\pi}{L}} \right) - \tan^{-1}\left(\frac{e^{\frac{-ns\pi}{L}} - \cos\frac{b\pi}{L}}{\sin\frac{b\pi}{L}} \right) \right] \quad . \; . \; (22)$$

Equation 22 is not necessary in the usual slab problem, for it is more convenient to use a very simple relationship which can be derived to express *division* of the total heat flow from the source between the two sides of the slab. From Equation 5,

$$\frac{m}{s} = (q/A)_1 + (q/A)_2$$

Substituting in Equation 22 and referring to surface 1,

$$\frac{\left(\frac{q}{A}\right)_1}{\left(\frac{q}{A}\right)_1 + \left(\frac{q}{A}\right)_2} = \frac{2}{\pi} \sum_{n=-\infty}^{+\infty} \left[\tan^{-1}\left(\frac{e^{\left(\frac{(s/2 - ns)\pi}{L}\right)} - \cos\frac{b_1\pi}{L}}{\sin\frac{b_1\pi}{L}} \right) - \tan^{-1}\left(\frac{e^{\left(\frac{-ns\pi}{L}\right)} - \cos\frac{b_1\pi}{L}}{\sin\frac{b_1\pi}{L}} \right) \right] \quad . \; . \; (23)$$

Equation 23 expresses the fraction of the total source heat flow passing through surface 1. To first show that this fraction is independent of s, evaluate its partial derivative with respect to s and demonstrate that this derivative has the value zero. The variable appears only in the exponential factors; the derivative is

$$\frac{\partial \left[\frac{\left(\frac{q}{A}\right)_1}{\left(\frac{q}{A}\right)_1 + \left(\frac{q}{A}\right)_2} \right]}{\partial s} = \frac{1}{L} \sum_{n=-\infty}^{+\infty} \left[\frac{\left(\frac{1-2n}{2}\right) \sin\frac{b_1\pi}{L}}{\cosh\left[\frac{s}{L} \cdot \frac{(1-2n)\pi}{2}\right] - \cos\frac{b_1\pi}{L}} + \frac{n \sin\frac{b_1\pi}{L}}{\cosh\frac{sn\pi}{L} - \cos\frac{b_1\pi}{L}} \right] \quad . \; . \; . \; (24)$$

To show that the summation is zero, a few terms may be written out.

$$\sum_{n=-\infty}^{+\infty}\left[\begin{matrix}\text{equation}\\(24)\end{matrix}\right] = \overbrace{\frac{\frac{1}{2}\sin\frac{b_1\pi}{L}}{\cosh\frac{s\pi}{2L}-\cos\frac{b_1\pi}{L}} + 0}^{n=0} - \overbrace{\frac{\frac{1}{2}\sin\frac{b_1\pi}{L}}{\cosh\frac{s\pi}{2L}-\cos\frac{b_1\pi}{L}} - \frac{\sin\frac{b_1\pi}{L}}{\cosh\frac{s\pi}{L}-\cos\frac{b_1\pi}{L}}}^{n=-1}$$

$$+ \overbrace{\frac{\frac{3}{2}\sin\frac{b_1\pi}{L}}{\cosh\frac{3s\pi}{2L}-\cos\frac{b_1\pi}{L}} + \frac{\sin\frac{b_1\pi}{L}}{\cosh\frac{s\pi}{L}-\cos\frac{b_1\pi}{L}}}^{n=+1} - \overbrace{\frac{\frac{3}{2}\sin\frac{b_1\pi}{L}}{\cosh\frac{3s\pi}{2L}-\cos\frac{b_1\pi}{L}} - \frac{2\sin\frac{b_1\pi}{L}}{\cosh\frac{2s\pi}{L}-\cos\frac{b_1\pi}{L}}}^{n=-2} + \cdots \quad (25)$$

In the terms written out, the only one uncancelled is the second term for $n = -2$, but this would be cancelled if the terms for $n = 2$ had been written. As the summation is extended to larger and larger values of n $(\pm)$, the remaining or uncancelled term approaches zero as a limit (because of the *cosh* factor approaching infinite magnitude). Hence, it may be concluded that the fraction of the total source heat flow passing through either slab surface is ***independent of the source spacing.*** This greatly simplifies application calculations.

The final analytical step is to establish the rule for the division of the total source heat flow to the two sides of the slab in terms of the remaining geometrical factors, the bury b and the slab thickness L.

Referring to Equation 23 for the division of the total source heat flow, since this division is independent of the tube spacing s it is permissible to consider (s/L) in Equation 23 to be a very, very large magnitude. Physically this establishes a condition where the heat flow over the range from $-s/2$ to $+s/2$ about a source is influenced only by the single source because the others are too far away. Mathematically, with s very large the two terms of the summation cancel each other for any $n \neq 0$, since both terms equal $\pi/2$ for negative n or both equal $\frac{(-\pi b_1)}{L}$ for positive n. The only factors remaining in Equation 23 are those for $n = 0$; but since s is very large the first term becomes $\pi/2$. Thus,

$$\frac{(q/A)_1}{(q/A)_1 + (q/A)_2} = \frac{2}{\pi}\left[\frac{\pi}{2} - \tan^{-1}\frac{e^0 - \cos\frac{b_1\pi}{L}}{\sin\frac{b_1\pi}{L}}\right]$$

$$= 1 - \frac{2}{\pi}\tan^{-1}\frac{1 - \cos\frac{b_1\pi}{L}}{\sin\frac{b_1\pi}{L}} \quad (26)$$

Making use of the trigonometric identity $\tan\frac{\Psi}{2} = \frac{1-\cos\Psi}{\sin\Psi}$, the expression becomes

$$\frac{(q/A)_1}{(q/A)_1 + (q/A)_2} = 1 - \frac{2}{\pi}\,\frac{\pi b_1}{2L} = 1 - \frac{b_1}{L} \quad (27)$$

The subscripts 1 and 2 in Equation 27 may be interchanged. Equation 27 establishes the valuable practical result that the division of the total *source* heat flow between the two sides of the slab varies linearly with the bury-to-thickness ratio. Test data agree with this conclusion.

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No. 1389

ELECTRIC ANALOGGER STUDIES ON PANELS WITH IMBEDDED TUBES†

By Carl F. Kayan*, New York, N. Y.

This paper is the result of research sponsored by The American Society of Heating and Ventilating Engineers in cooperation with Columbia University.

Radiant heating, by means of heated imbedded tubes in concrete building slabs having top and bottom exposed surfaces, involves an upward and a downward heat flow to the realm of lower ambient space temperature. When heating tubes are thus imbedded within a slab structure, they set up complex heat-flow paths for which prediction of the slab temperatures and the resultant heat transfer is difficult.

With the flow of a heating medium through the imbedded tubes, the temperature of the slab surfaces is increased above that of the ambient space. Thus the operating temperature-difference effect for the surfaces is created. With asymmetrical tube location, and in addition, with asymmetrical thermal-resistance conditions above and below the slab, the upper and the lower surfaces constitute, to differing extent, the effective heat transfer areas for the slab.

Particularly under these circumstances, the calculation of the heat-flow distribution between the two surfaces is troublesome. The present analysis is predicated on the inherent two-way heat-flow effect, and has been carried out for two slab structures with various active tube spacings, by means of the *Electric Analogger.*

This investigation, by electrical analogy method, has been undertaken as part of the program of the A.S.H.V.E. Technical Advisory Committee on Panel Heating and Cooling, *Group A.*

In addition to the objective of determining heat flow and temperature distribution in this study, one particular aspect has been the determination of the

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† Abstract of this report presented at the 56th Annual Meeting of The American Society of Heating and Ventilating Engineers, Dallas, Tex., January 1950.

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comparative effect of *idle* tubes in test panels such as tested by the A.S.H.V.E. Research Laboratory[1] where tubes were cast into the slab on 4 in. centers, and tests were made with different combinations of active and idle tubes. The presence of the idle tubes posed a question as to their relative influence on the overall results.

Fig. 1 shows the structural arrangements investigated in the present program. Section (*a*) shows a monolithic concrete panel with ¾ in. tubes on 4 in. centers and bottom cover of 5 in. The top cover is 3½ in. (Series A tests). Section (*b*) shows a panel with ¾ in. top cover (Series B tests).

For all of the investigations in this study, the conductivity of the concrete has been taken at $k = 9.00$ Btu per (sq ft) (hr) (F deg per in.), and the values

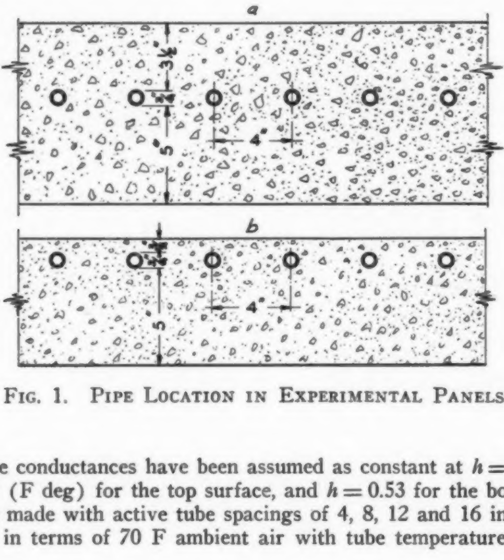


FIG. 1. PIPE LOCATION IN EXPERIMENTAL PANELS

of the surface conductances have been assumed as constant at $h = 1.11$ Btu per (sq ft) (hr) (F deg) for the top surface, and $h = 0.53$ for the bottom surface. Studies were made with active tube spacings of 4, 8, 12 and 16 in., and results are reported in terms of 70 F ambient air with tube temperatures of 120 and 170 F.

EXPERIMENTAL MODEL RELATIONSHIPS

In accordance with previously described procedures which need not be detailed here[2, 3, 4], and using electrically conductive flat sheet, a full-scale 48-in. section *Analogger* model was constructed to represent the two-dimensional heat flow conditions of the panels. This is shown in Fig. 2, with electrical connections. Two-way slab heat flow was simulated by two-way electrical flow. Electrodes, to represent the ambient-air conditions above and below, are shown in the diagram, along with the electrodes to represent the imbedded tubes. In the test program, the active-tube electrodes were connected to the source of electrical

[1] Exponent numerals refer to References.

current, with the inactive *idle* tubes disconnected but left in place on the model. To determine the comparative effect, due to the presence of these idle tubes in the direct experiments of the A.S.H.V.E. Research Laboratory, separate tests were made in the ¾ in. top-cover series with the idle-tube electrodes physically removed from the model. The ¾ in. top-cover series was regarded as the more critical in its idle tube aspect.

The electrical analogy procedure is based on the similarity between the flow of heat in thermal circuits and the flow of current in electrical circuits, *i.e.*, Ohm's law for electrical resistance circuits has its counterpart in heat flow circuits. This is conventionally identified as the *resistance concept* of heat transfer.[2] In accordance with this principle, the characteristics of the heat flow

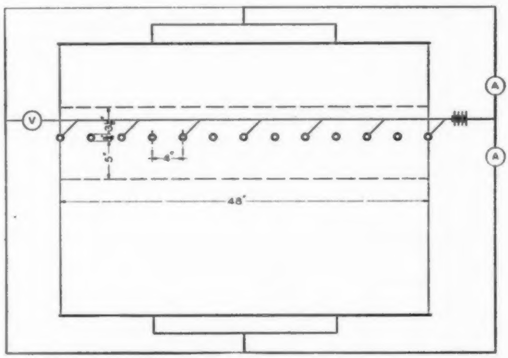
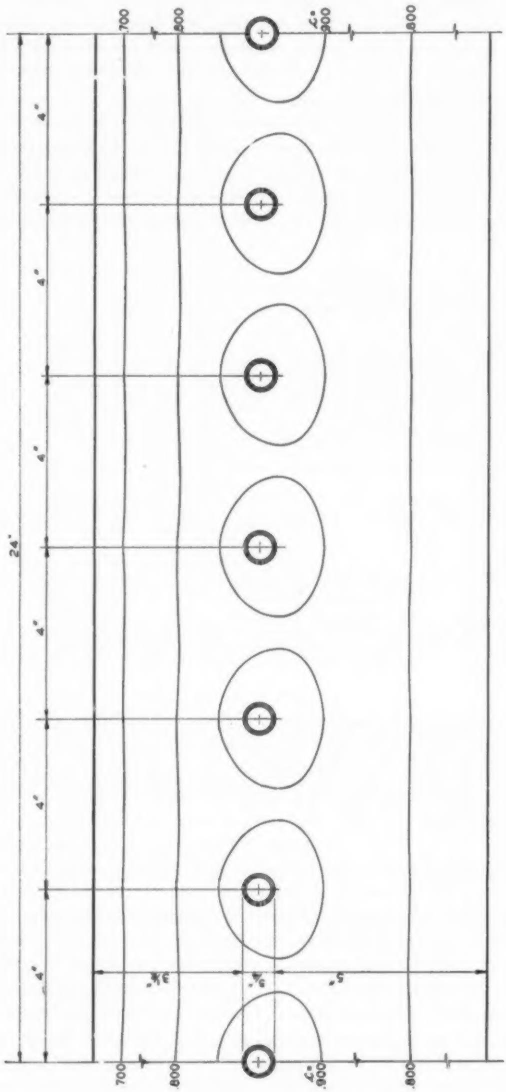


FIG. 2. DIAGRAM OF 48-INCH ANALOGGER MODEL

are established, by analogy, through the study of an equivalent electrical system. Determination of the isopotential lines through the electrical study then permits establishment of the temperature distribution in accordance with the existing overall temperature difference for the given thermal structure.

In the present *Analogger* model, regions between the electrodes on the conductive sheet represent, to scale, the electrical resistance sections equivalent to the thermal resistance sections. The effect of the boundary conductance at the surfaces is introduced by off-setting the *ambient-air* electrodes a given distance from the panel section, commensurate with the thermal resistance. (Thus for example, for $h = 0.53$ and $k = 9.00$, x_e will be 17 in., as explained later in text referring to Equation 6.) On the sheet model, these sections for the top and bottom conditions were, in addition, slit at regular intervals, perpendicularly to the electrodes, to prevent lateral electrical flow in the *boundary* region.

At the very outset it must be emphasized that the results of the present study are based solely on the assumed basic thermal conductivity and boundary conductances. These values, though representative, have been selected arbitrarily and are, in addition, taken as independent of temperature. The joint between the tubes and the concrete is presumed to represent negligible resistance.



(Series A, 3½ in. top cover)

FIG. 3. ISOPOTENTIAL PATTERNS FOR 4-INCH TUBE SPACING

Electrical measurements on the model yield the potential difference ratio c, defined:

$$c = \Delta e_p / \Delta e. \quad (1)$$

where

Δe_p = potential difference between the ambient-air electrodes and a given point p within the panel section;

Δe = electrical potential difference between the active tube electrodes (equi-potential) and the air-electrodes (equi-potential).

The value of c is most conveniently established by means of the slide-wire bridge circuit.[8]

Equivalent temperatures throughout the slab may be established on the basis of the overall temperature conditions:

$$t_p = t_a + c\Delta t \quad (2)$$

where

t_p = Fahrenheit temperature at a given point p.

t_a = Fahrenheit temperature of the ambient air.

Δt = overall temperature difference between the imbedded tubes and the air, degrees.

The following thermal resistance relationships are of interest:

$$R_a = 1/h_a \quad (3)$$

where

R_a = equivalent thermal resistance at the air boundary, Fahrenheit degrees, Btu per (square foot) (hour).

h_a = air boundary surface conductance, Btu per (square foot) (hour) (Fahrenheit degree).

The resistance R_w of wall or slab material of uniform composition:

$$R_w = x_w / k \quad (4)$$

where

x_w = thickness of wall or slab, inches.

k = thermal conductivity of wall or slab material, in Btu per (square foot) (hour) (Fahrenheit degrees per inch).

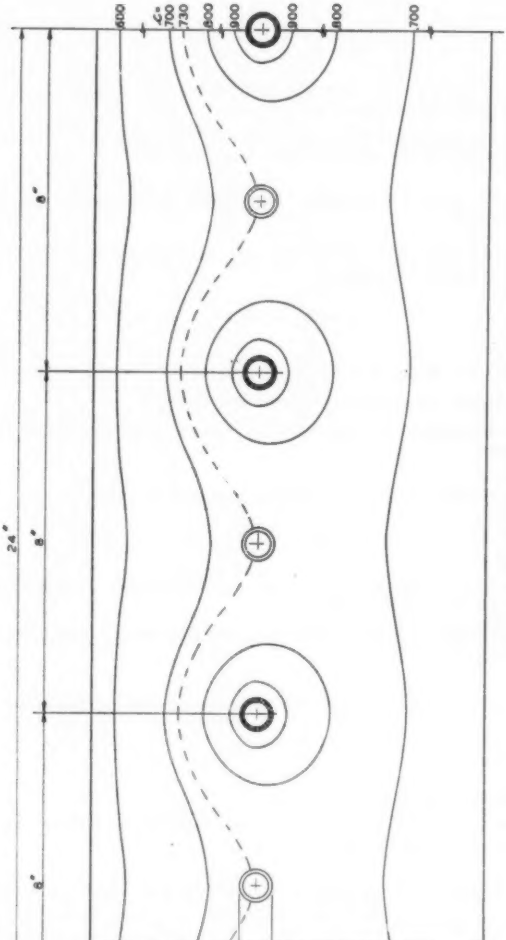
As previously indicated in Equation 3, for a given value of h_a there is a corresponding value of resistance R_a. Thus, for a given conductivity of slab material, there is some equivalent thickness x_e of slab material that would give the same resistance R_e to heat transfer as the air boundary:

$$R_a = 1/h_a = R_e = x_e/k \quad (5)$$

Thus,

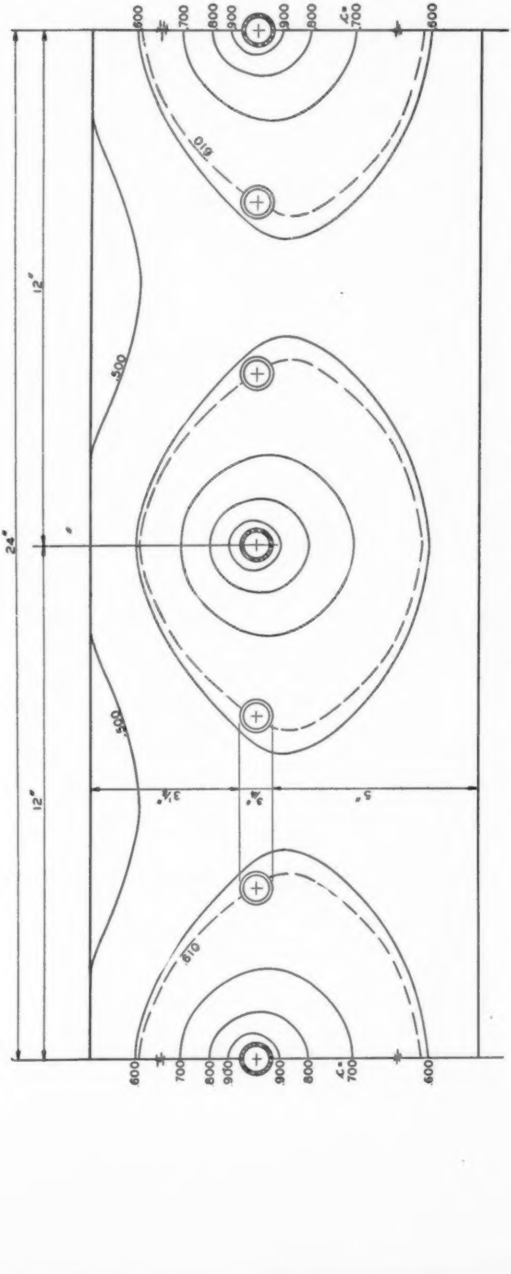
$$x_e = k/h_a \quad (6)$$

This value of x_e is employed in the construction of the Analogger model, and represents the distance on the full-scale model by which the air electrodes are



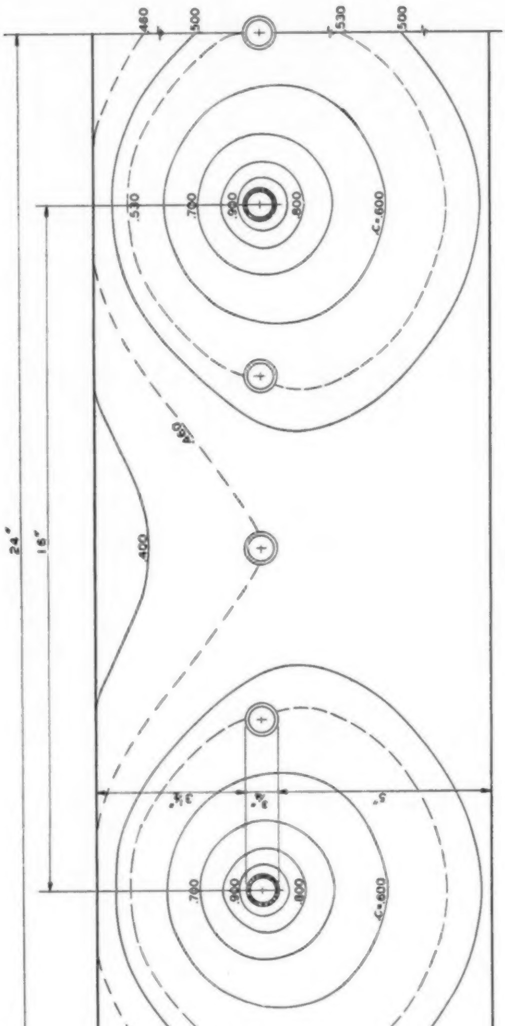
(Series A, 3½ in. top cover)

FIG. 4. ISOPOTENTIAL PATTERNS FOR 8-INCH TUBE SPACING



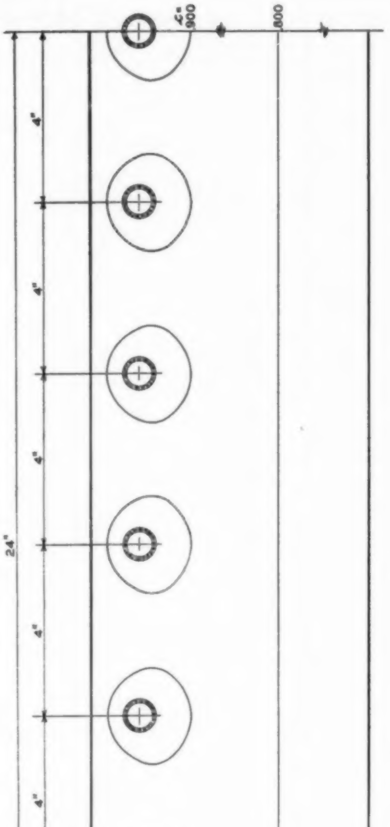
(Series A, 3½ in. top cover)

FIG. 5. ISOPOTENTIAL PATTERNS FOR 12-INCH TUBE SPACING



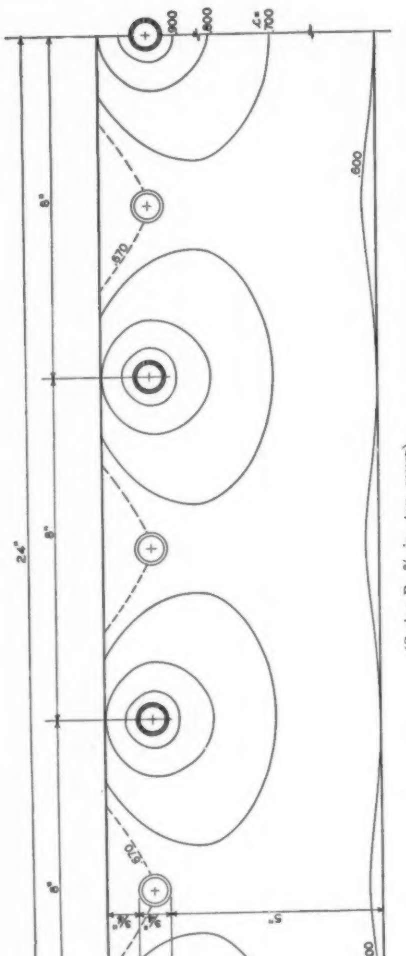
(Series A, 3½ in. top cover)

FIG. 6. ISOPOTENTIAL PATTERNS FOR 16-INCH TUBE SPACING

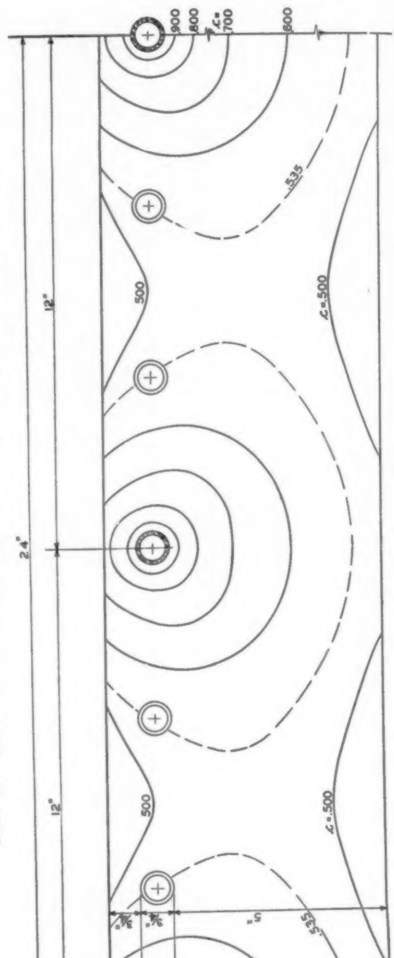


(Series B, ¾ in. top cover)

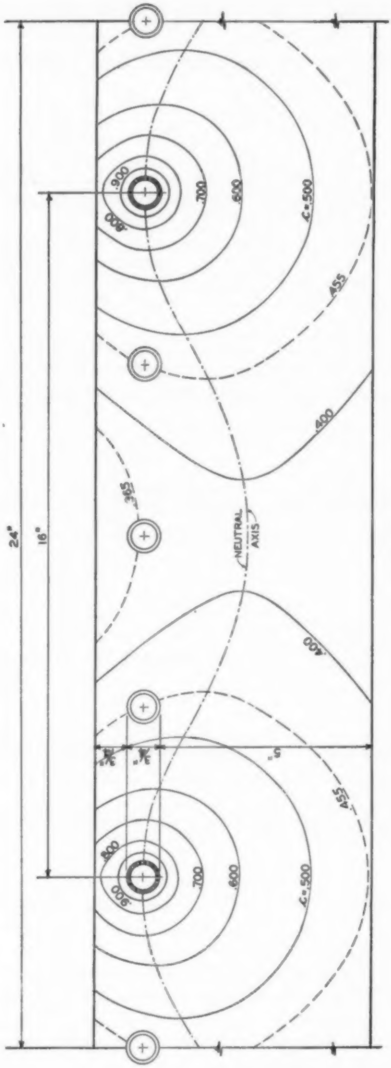
Fig. 7. Isopotential Patterns for 4-Inch Tube Spacing



(Series B, ¾ in. top cover)
FIG. 8. ISOPOTENTIAL PATTERNS FOR 8-INCH TUBE SPACING



(Series B, ¾ in. top cover)
FIG. 9. ISOPOTENTIAL PATTERNS FOR 12-INCH TUBE SPACING



(Series B, ¾ in. top cover)

FIG. 10. ISOPOTENTIAL PATTERNS FOR 16-INCH TUBE SPACING

off-set from the panel section. Thus, distances on the model produce electrical-resistance values having the same relationship to one another as the thermal resistances, *i.e.*, linear distances on the conductive sheet are commensurate with thermal resistances.

Electrical measurements also make possible the determination of an overall resistance ratio f:

$$f = r_t/r_w \qquad (7)$$

where

r_t = measured overall electrical resistance on the model between an air-electrode and the tube electrodes, based on current and voltage values, under the different test conditions;

r_w = electrical resistance of the model reference strip representing the plain slab, *i.e.*, the resistance of sheet material as measured between two parallel electrodes spaced apart by a distance equal to the slab thickness, and extending the full width of the model (48 in. in this case).

This dimensionless electrical ratio f has its counterpart in an equivalent thermal ratio F_T, and represents the transition bridge between the measured electrical flow for the model and the predicted heat flow for the slabs. Thus

$$f = F_T = R_t/R_w \qquad (8)$$

where

R_t = thermal resistance (per square foot, average, for the 48 in. slab section) of the slab-tube assembly under the different conditions;

R_w = thermal resistance (per square foot) of the reference thickness for the slab, *i.e.* $= x_w/k$.

Thus

$$R_t = R_w F_T \qquad (9)$$

It should further be noted for heat-flow computation:

$$H_{aver} = \Delta t/R_t \qquad (10)$$

where

H_{aver} = average heat flux for the slab surface, Btu per (square foot) (hour);

Δt = overall temperature difference between air and tubes.

For each surface of area A, in square feet, the gross heat flow q, in Btu per hour is given by

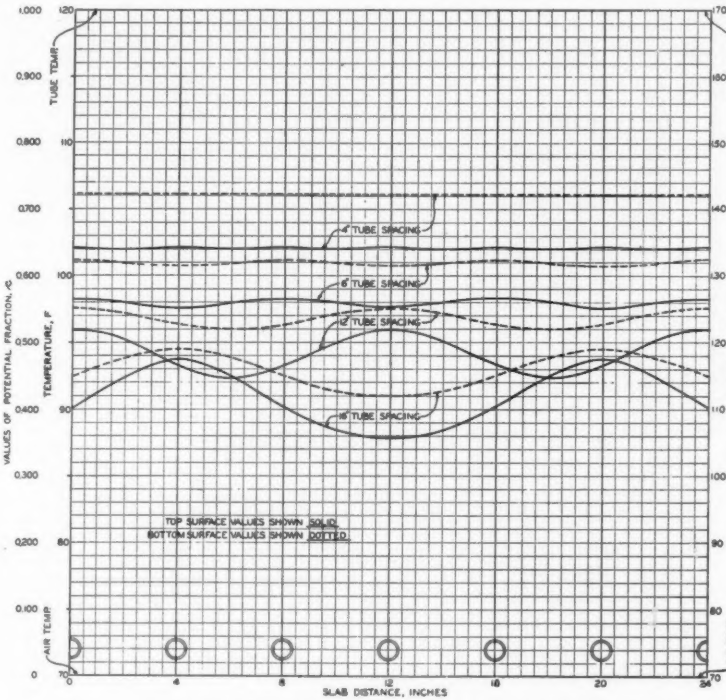
$$q = AH_{aver} \qquad (11)$$

TEST PROGRAM

As noted earlier, this investigation comprised studies on two concrete slab constructions shown in Fig. 1, each with ¾ in. tubes on 4 in. centers and 5 in. bottom cover, as follows: Series A, top cover of 3½ in.; Series B, top cover of ¾ in. The extent of the slab studied was 48 in. and the *Analogger* model was made to scale with a width of 48 in.

For both series, electrical studies for isopotential patterns and resistance values were made with different tube combinations. One test covered conditions

with all the tubes active, *i.e.*, active tubes on 4 in. centers. Another test covered active tubes on 8 in. centers, with the single intermediate tubes idle remaining in place on the model but *unheated, i.e.*, disconnected. In another test, active tubes on 12 in. centers were used with two intermediate idle tubes, and in another test, active tubes on 16 in. centers, with three intermediate idle tubes were used.



Top and bottom temperatures shown for 170 and 120 F water temperature

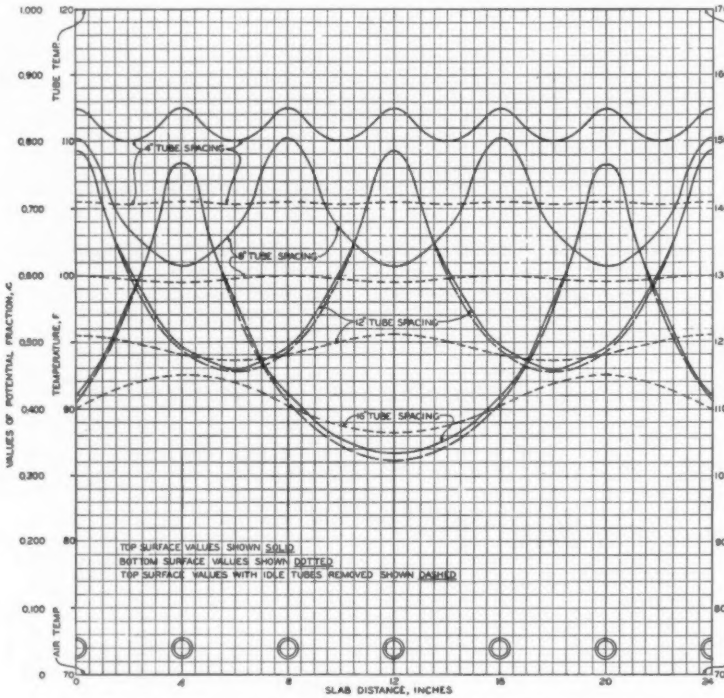
FIG. 11. PANEL SURFACE TEMPERATURE VARIATION FOR 3½-INCH COVER

Resistance studies were made in all these tests, and also, in addition, on 24 in. active tube spacing.

Although the regularity of the temperature patterns for the 3½ in. top-cover series indicated that the idle tubes exerted no particular influence, even with a low conductivity value ($k = 9.00$) for concrete, the idle-tube aspect was further probed under the more severe conditions of the thin top cover of ¾ in. For this case, the idle tube *electrodes* were actually removed from the set-up, and both temperature-pattern and resistance studies carried on, for comparison with the conditions in which the idle tubes remained in place but were electrically disconnected.

TEMPERATURE RESULTS

For all of the studies, the air temperature was taken at 70 F, and the results evaluated in terms of two tube temperatures, namely, 120 F and 170 F. As noted previously, temperatures may be derived from the values for the potential



Top and bottom temperatures shown for 170 and 120 F water temperature

FIG. 12. PANEL SURFACE TEMPERATURE VARIATION FOR ¾-INCH COVER

difference ratio *c*. Table 1 permits ready translation of values of *c* (in isopotential patterns) into equivalent temperatures for the slab under the two tube-temperature conditions.

Figs. 3, 4, 5 and 6 show the isopotential patterns for Series A (3½ in. top cover), with 4 in., 8 in., 12 in. and 16 in. active tube spacings respectively, and with the intervening tubes idle as previously indicated. The comparative effect of the spacing on the location of the different isotherms may be seen directly. The regularity of the equivalent temperature patterns, in spite of the idle tubes, is also to be noted, indicating little, if any, effect from the idle tubes. The isotherms going through the idle tubes are also shown.

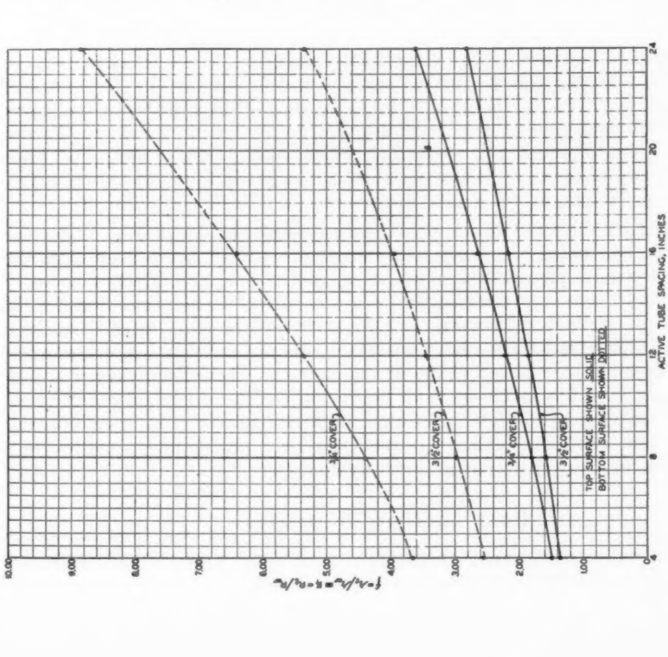


FIG. 13. VALUES OF $f = F_T$ IN RELATION TO ACTIVE TUBE SPACING

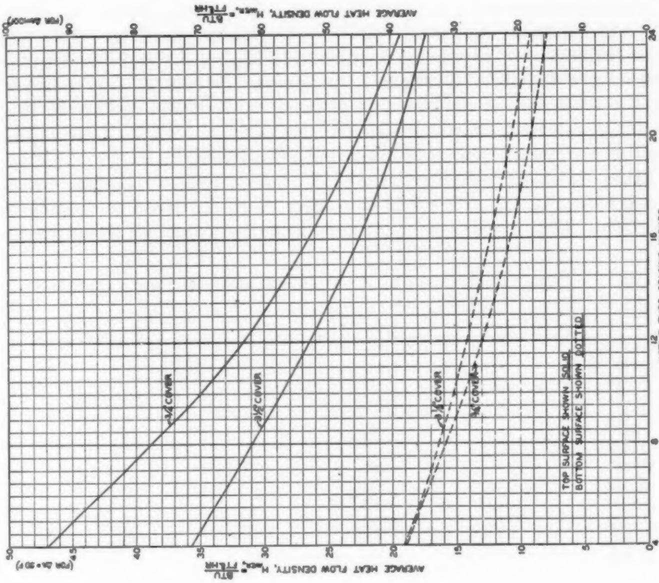


FIG. 14. AVERAGE HEAT FLOW PER SQUARE FOOT FROM TOP AND BOTTOM SURFACE FOR 120 AND 170 F WATER TEMPERATURE

Figs. 7, 8, 9 and 10 show the patterns for Series B, with the ¾ in. top cover, for the 4, 8, 12, and 16 in. active tube spacings respectively, and corresponding to the Series A patterns of Figs. 3, 4, 5, and 6. Here, also, the paths of the idle-tube isotherms are indicated throughout the panel. A neutral axis typically illustrating the separation of the slab into zones of upward and downward heat flow components has been added to Fig. 10.

Of particular interest are the temperature variations along the top and bottom surfaces. These are shown for Series A, 3½ in. cover, in Fig. 11, and for Series B, ¾ in. cover, in Fig. 12. For both, the effect of the active tube spacing is markedly evident. The spread of temperatures, from the maximum at the active-tube location to the minimum at the midpoint between the active tubes, is to be noted. The variation for the bottom surface is considerably less than for the top surface, by virtue of the greater effective thermal resistance on the lower section. The comparison between the 3½ and the ¾ in. top covers is likewise noteworthy, high-lighting the controlling influence of a thin cover. For Figs. 11 and 12, two temperature scales are shown: the left is for a tube temperature of 120 F, and the right, for a tube temperature of 170 F.

TABLE 1—SLAB TEMPERATURE CORRESPONDING TO POTENTIAL DIFFERENCE RATIO C

POTENTIAL DIFFERENCE RATIO c	TUBE TEMPERATURE	
	120 F	170 F
1.000	120	170
0.900	115	160
0.800	110	150
0.700	105	140
0.600	100	130
0.500	95	120
0.400	90	110
0.300	85	100

The effect of idle tubes in the slab is directly brought out in Fig. 12 for the ¾ in. cover. As noted previously, separate pattern studies were made with the idle-tube electrodes physically removed from the model. The effect is not very great, as can be seen from the comparison of top-surface temperature lines for the 12 and 16-in. spacings. However, the irregularity of the temperature lines with idle tubes in place is to be noted, particularly the *flat* sections where the intermediate idle tubes have distorted the normal pattern. This effect was not noticeable with the thick 3½ in. cover. With the removal of the idle tubes, the temperature lines became more regular, and were slightly displaced from their previous position. The idle-tube influence proved relatively negligible on the overall-resistance values: differences could scarcely be noticed.

HEAT TRANSFER EVALUATIONS

Through the determination of electrical resistance values from the model studies, the dimensionless resistance ratio f of Equation 7 was established. For

Series A, the value of r_w was taken in terms of the slab thickness of 9.25 in., and for Series B, in terms of the corresponding slab thickness of 6.50 in. The overall resistance for the different tube combinations was determined separately for both the top-cover and bottom-cover portions of the slab, with the idle tubes in place on the model. As previously indicated, the effect of removing the idle tubes from the model proved scarcely noticeable.

Fig. 13 shows the value of $f = F_T$ plotted against active-tube spacing, for both top and bottom portions of the two slabs under study.

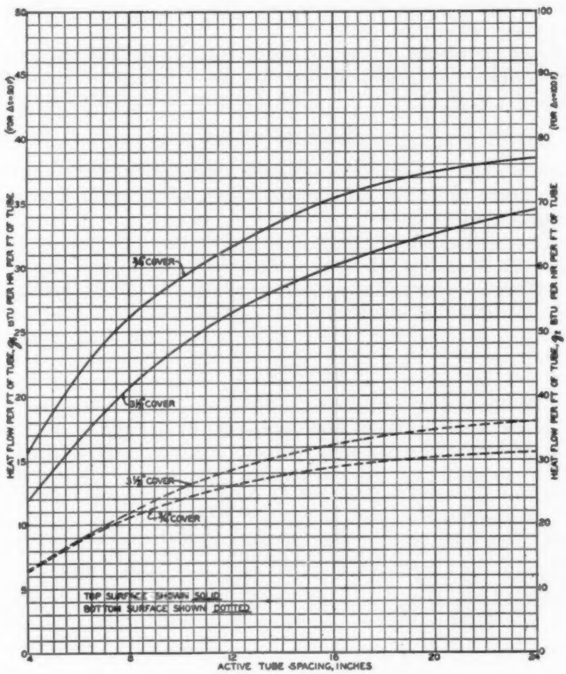


FIG. 15. HEAT FLOW PER FOOT OF TUBE LENGTH FOR 120 AND 170 F WATER TEMPERATURE

In Fig. 14, through the use of ratio F_T and the corresponding value for $R_t = R_w F_T$ (Equation 9), the value of H_{aver} in Btu per (sq ft) (hour) has been plotted against tube spacing (see Equation 10). Two scales are shown, the left one for tube temperature of 120 F, and the right one for tube temperature of 170 F. To correspond with the electrical evaluation, R_w has been based on the actual slab thickness, *i.e.*, 9.25 in. for Series A, and 6.50 in. for Series B.

From the results shown in Fig. 14, the heat transfer performance for the two slabs may be compared directly. It should be remembered that the two slabs are

alike in all respects, thermal and structural, except for the thickness of the top cover, *i.e.,* 3½ in. *vs.* ¾ in. The increased top-surface values of H_{aver} for the ¾ in. cover over those for 3½ in. cover are readily to be noted and easily accounted for. Particular attention is directed, however, to the performance of the bottom surfaces, for which the two slab conditions were presumably identical. As a result, however, of the decreased top cover of ¾ in. instead of 3½ in., the heat flux through the bottom was directly influenced, as may be seen from the decreased values for the ¾ in. cover curve. It may thus be recognized that the differing distortion of the isotherms through the bottom cover is directly associated with the thickness of the top cover. This is likewise evident from the study of the slab temperature patterns.

Fig. 15 shows the results of Fig. 14 converted to heat flow values per foot of tube length, typical of necessary design data for the output of panel surface.

ACKNOWLEDGMENT

The author wishes to acknowledge, with appreciation, the support of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in making this cooperative project possible.

Acknowledgment is also gratefully made to C. M. Humphreys, senior engineer, A.S.H.V.E. Research Laboratory, Cleveland, for helpful suggestions, and to Prof. J. Arthur Balmford, Electrical Engineering Department, Columbia University, New York, for his cooperation on electrical details.

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1390

No. 1390

EFFECT OF PANEL LOCATION ON SKIN AND CLOTHING SURFACE TEMPERATURE

By L. P. Herrington* and R. J. Lorenzi**, New Haven, Conn.

In the past few years low temperature radiant heating panels have seen increasing use in this country. A number of publications have considered the problem of the most advantageous location of such panels, with emphasis on installation and constructional features.[1,2] Recent studies at this laboratory of the performance of electrically heated panel systems have directed our attention to the comfort aspects of this problem. It seems probable that under certain conditions of room geometry, installation procedure, and heating load, panels installed in ceiling, floor, or side walls may give satisfactory heating results. However, under average conditions of installation, the various locations may have substantially different possibilities for discomfort.

The primary difference in the effect upon the occupant of different panel locations would appear to be a directional heating effect. This effect should show itself in variations in the regional surface temperatures of the skin and clothing of the occupants. Floor panels may overheat the lower segment of the body, ceiling panels may produce overheating of the head, and side panels could conceivably produce unilateral, and undesirable, differences in occupant surface temperature. Since comfort and occupant surface temperatures are closely related, experiments recording these variables under comparable conditions with floor and ceiling panel locations have been conducted.

Experimental Test Room

The experimental work was conducted in the climatic test rooms of the Pierce Laboratory of Hygiene, whose construction has been described elsewhere.[3] In brief resumé, these consist of two rooms 15 ft by 12 ft by 8 ft high, separated by an instrument corridor 4 ft wide. Three windows opening on the climate controlled shell spaces provide, in conjunction with painted plaster walls, oak floors, and side bracket lighting, the appearance of a small living room. The

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1 Exponent numerals refer to References.

Presented at the 56th Annual Meeting of The American Society of Heating and Ventilating Engineers, Dallas, Tex., January 1950.

entire wood framed structure is enclosed within a larger climate conditioned shell space, so divided as to permit separate temperature and humidity control over a wide range in the space adjacent to each of the 6 sides of the building cube. The value of *U*, the coefficient of heat transmission for the wall structure was 0.25 Btu per (hour) (sq ft) (F deg).

Test room No. 1, used for these studies, was furnished with four comfortable but unupholstered arm chairs. Two of these were placed centrally against the inner wall and faced tables used by the observers. Two chairs used for subjects were placed so that each was at the center of a rectangle 12 x 7.5 ft, representing one half of the floor space. A low side table was placed near each chair on the inner wall side for reading matter and record forms.

Temperatures of floor and ceiling surfaces were obtained from recorders showing the average voltage of six thermocouples on each surface. Air temperatures were recorded at room center by a shielded thermocouple. Mean black body temperature at room center was obtained from a thermocouple placed in the center of a blackened globe thermometer, and this room center black body temperature was the temperature held constant in the experiments with different panel locations. Skin and clothing surface temperatures were measured by a multiple junction thermopile of the Hardy type.

Since the problem under test was the directional effect of a stable equilibrium room surface temperature on occupant surface temperature, it was possible to use the basic equipment of the climatic unit to produce normally heated floors or ceilings. The shell spaces adjacent to five of the six surfaces of the test room could be maintained individually at any temperature between 0 F and 120 F. Hence, by operating the side spaces at outdoor temperatures, and either the ceiling or floor shell space at an elevated temperature, floor or ceiling panel heat could be provided.

OBSERVATIONAL PLAN

Two young men were used as subjects. They were clothed in light underwear, shirt and tie, light socks, shoes, and a suit of moderate weight without vest. Ten days of observation divided into two periods of five days with ceiling heating, and five days with floor heating were scheduled. In all experiments the primary condition of comparability was the 36 in. globe thermometer at room center.[4] Conditions were controlled so that this reading of combined radiant and air temperature effects was very near 75 F for all experiments. Shell spaces representing outside climate were set at 35 F. Under these conditions the standard globe thermometer control temperature of 75 F was obtained, with floor surface temperatures ranging from 78 to 80 F, and with ceiling temperatures ranging from 94 to 95 F.

Subjects entered the test room between 1 and 2 p.m. after 30 min of quiet reading in the main laboratory. They were seated in the centrally positioned chairs, and remained there for a two-hour period. Reading, smoking, and talking were permitted. The following data were collected for each two-hour period:

1. 36 in. room center black globe temperature (mean air-radiant temperature—to be standard at 75 F, globe reading)
2. 36 in. room center, air temperature
3. Mean floor surface temperature
4. Mean ceiling surface temperature

TABLE 1—MEAN VALUES FOR ENVIRONMENTAL TEMPERATURES, SUBJECT SURFACE TEMPERATURES, AND COMFORT VOTES IN EAST TEST ROOM USING UNIFORMLY HEATED FLOOR, F

(Mean Ceiling Temperature, All Tests, 70.0 ± 0.5 deg)

Test No.	Black Body Temp Room Center	36 In. Air Temp	Mean Floor Temp	Lower Extremities Temp	Head Temp	Mean Exposed Skin and Cloth Temp	Mean Comfort Vote	Mean Temp Vote
1	75.4	75.0	79.6	85.7	91.2	85.6	4.0	3.7
2	75.0	74.3	77.9	84.1	90.2	84.3	4.0	2.0
3	74.3	73.5	78.2	83.4	89.8	83.4	3.3	2.3
4	75.0	74.0	78.5	84.5	90.2	84.8	3.3	2.3
5	75.1	74.5	78.5	84.9	90.3	85.0	3.2	2.7
Mean Value	74.96	74.26	78.54	84.52	90.34	84.62	3.56	2.60

5. Average surface temperature of the lower extremities. Thermopile readings at 4 points on clothing and shoe surface between thigh level and foot level

6. Average head surface temperature. Thermopile readings at three points; cheek, upper hair surface, and dorsal neck surface

7. Mean exposed skin and clothing surface temperature. Composed of (5), (6), plus one thermopile reading of hand temperature, two readings on sleeve surface, and four readings divided between front and back areas of the trunk. Mean skin temperature readings determined by weighting the foregoing values for relative area (head, 7 percent; upper extremities, 21 percent; trunk 31 percent; and lower extremities, 41 percent)

8. Mean comfort vote, obtained by subject vote after one and two hours of quiet, normally clothed exposure to the heating condition, and recorded on the following scale:

VOTE

1. Uncomfortably cold
2. Comfortably cool
3. Comfortable
4. Comfortably warm
5. Uncomfortably hot

9. Mean temperature vote. Subject appreciation of the terms *comfort* and *temperature* are not always identical. At the end of 45 min and 90 min, in the two hour exposure, subjects voted on a 3-point scale:

VOTE

1. Cool
2. Indifferent
3. Warm

RESULTS

In Table 1 the mean values for five days of observation on the significant room temperatures, subject surface temperatures, and comfort votes are given for the uniform heat floor panel condition.

TABLE 2—DATA COMPARABLE TO TABLE 1 FOR UNIFORMLY HEATED CEILING, F

(Mean Floor Temperature, All Tests, 72.8 ± 0.5 deg)

Test No.	Black Body Temp Room Center	36 In. Air Temp	Mean Ceiling Temp	Lower Extremities Temp	Head Temp	Mean Exposed Skin and Cloth Temp	Mean Comfort Vote	Mean Temp Vote
11	75.8	73.4	95.1	82.3	89.8	83.7	2.7	2.0
12	75.1	74.3	95.7	82.5	91.5	84.0	3.3	2.0
13	75.0	74.3	94.2	82.9	91.7	85.3	3.3	2.3
14	74.9	72.7	94.6	82.4	90.5	83.6	3.4	2.2
15	75.5	74.5	95.1	83.6	92.4	84.2	3.0	2.0
Mean Value	75.26	73.84	94.94	82.74	91.18	84.16	3.14	2.10

In Table 2 similar data for five tests with uniform ceiling heating are given. By the use of long periods of pre-test equilibration, the desired mean air-radiant temperature at the 36 in. level (room center) was held within the limits of 74.3 to 75.4 F. Floor temperatures averaged near 79 F, and ceiling temperatures very near 95 F, in the respective series for floor and ceiling radiant heating. Mean exposed skin and clothing temperature was nearly identical for the two conditions of heating. The heated floor series showed a mean subject surface temperature value only 0.5 deg different from the ceiling series. In both series, extreme values for individual tests differed by 2 deg with the variation around the mean being approximately 1 F ±.

Since conclusions are appropriately based upon mean values, the close agreement of both mean room center globe temperature and mean subject surface temperature warrants the statement that in both heating variations the conditions were comparable in an overall thermal sense.

LOCAL SURFACE TEMPERATURE OF SUBJECTS

When the head and lower extremity temperatures are examined, it is immediately apparent that substantial differences exist between the ceiling and floor heating with respect to local temperatures, despite closely comparable mean values for subject surface. When floor heating is used, the lower extremities are nearly 2 F warmer than when ceiling heating is used. In similar fashion head temperatures in ceiling heating series exceed those in the floor series by 0.8 to 0.9 F.

The differences are not large in an absolute sense; however, if we consider these differences in relation to the gradient between room temperature and the body surface, which is the proper method, the variation in clothing surface temperature of the lower extremities is considerable (as noted in Table 3).

From Table 3 it is apparent that when the total room center globe thermometer heating effect is comparable in two tests, provision of heat by a uniformly heated floor panel increases the gradient between lower extremities and the mean environmental temperature by approximately 28 percent as compared with the ceiling heating conditions. This means that in floor heating a larger proportion of heat is being lost to the environment through the legs and this may

reasonably b~ attributed to the dilating effect of higher contact floor temperatures on the blood vessels of the lower extremities.

In contrast to this, it is seen from Table 3 that the location of a heating source in the ceiling has produced an increase in head to environment gradient of less than 5 percent as compared with the gradient observed for the floor heating conditions.

While these results are not greatly surprising, they are worth emphasizing since the authors have not found any published data on this point. It is quite clear that a much larger change in the relative heat loss from legs and head is produced by floor heating than by ceiling heating.

COMFORT AND TEMPERATURE VOTES

The foregoing objective data are confirmed by the mean results on comfort and temperature votes. Although the total heating effect of both environments is almost exactly equal in a physical sense, the subjects' mean vote classified the floor heating conditions as comfortably warm in two tests, and in three others varied their votes between comfortable and comfortably warm.

In the ceiling heating series the mean vote indicated substantially less frequent votes of comfortably warm. The separate votes taken on the 3-step temperature sensation scale as a check (without mention of comfort) confirmed the foregoing observations.

DISCUSSION

The ideal indoor conditions of environmental temperature, humidity, and air movement are probably realized for most individuals living in the temperate zones during the relatively short transition periods between spring and summer, and summer and fall. During these periods, external temperatures of the order

TABLE 3—PERCENTAGE CHANGE IN TEMPERATURE GRADIENT BETWEEN HEAD AND LOWER EXTREMITIES AND 36 IN. MEAN AIR-RADIANT TEMPERATURE

Temperature Produced by Ceiling dnd Floor Heating in Tests in which the Total Heating Effect is Comparable

	CEILING HEATING, F		FLOOR HEATING, F	DIFFERENCE
36 in. Mean Air-Radiant temperature	75.26		74.96	
Mean lower extremity temperature	82.74	Percent increase in lower extremity temperature gradient produced by floor heating as compared with ceiling heating + 27.8 percent	84.52	
Gradient	7.48		9.56	2.08
Mean Head Temperature	91.18	Percent increase in head temperature gradient produced by ceiling heating as compared with floor heating + 3.5 percent	90.34	
Gradient	15.92		15.38	0.54

of 65 to 70 F require little or no positive heating. Sun effects and minor indoor loads result in indoor air and wall temperatures of the order of 70-72 F. This small indoor-outdoor thermal gradient, in conjunction with open windows and mild outside wind effect, results in a general indoor atmospheric environment of very pleasant character, and one probably unequalled by any practical heating system during the cold months.

In this laboratory the general physiological criteria of thermal comfort have been described in many experimental studies.[5] Data drawn from the representative article cited here, and from the survey article on skin temperature and human heat regulation by Sheard[6] make it quite apparent that one of the primary conditions of thermal comfort is a skin temperature ranging from approximately 80 F on the toes and sole of the foot to approximately 95 F on the trunk and certain facial areas, with an overall average for the skin surface of 90-92 F.

The low temperatures on the feet are the net result of (1) a large surface-small volume factor, and (2) the practical circulatory necessity of high vascular tone in the lower extremities as an anti-gravity adjustment favoring competent circulation in the more vital body regions. One of the conditions of an alert subjective state is the maintenance of this vascular tone, and in consequence, foot temperatures are very considerably below the general skin average. The reality of this relation of tone in the lower extremities to their temperature is demonstrated very convincingly by Sheard[6] in relation to the sudden rise in lower extremity temperature with anesthesia and on falling asleep. Since psychic relaxation under conditions which require alertness is closely akin to certain sensations of dullness and fatigue, it appears desirable, in heating procedures, to avoid local heating effects which may induce low vascular tone in the foot and leg.

From this standpoint the data here reported are regarded as sounding a note of caution in the use of floor heating systems. In England floor temperatures above 75 F were seldom required for floor heating installations, and temperatures above this would probably not be tolerated.

During a recent spring day of ideal indoor comfort, the following temperature relations were recorded in this laboratory (subject had been seated quietly for several hours).

Outdoor Air Temperature	68.5 F
Mean Wall Temperature	72.0
Ceiling Temperature	73.5
Floor Temperature	71.1
Air Temperature 30 in. level	70.6
Air Temperature 66 in. level	72.5
Air Temperature 120 in. level	73.3
Outside shoe surface	73.9
Ventral skin of foot	87.7
Lower surface of toes	84.6
Skin surface of calf	91.1
Skin surface of thigh	92.5
Skin of hand and fingers	93.1
Trunk surface	95.0
Cheek	95.8

It is obvious from the foregoing data that outside shoe temperatures are 10 deg or more below foot surface temperature. It is also clear that with floor

temperatures of the order of 80 F, the skin surface temperature of the foot must rise considerably above the range of 78 F to 85 F that has been observed for a group of individuals with comfortable conditions and floor temperatures of the order of 70 F.

The data of this experimental study plus the foregoing observations make it highly probable that (*a*) floor temperatures above 75 F are physiologically undesirable, and that (*b*) 80 F can be taken as the level at which the local effect of floor heat is clearly demonstrable in the lower extremities when contrasted with ceiling location for the radiant panel.

CONCLUSIONS

1. At a room center black body temperature of 75 F, radiant floor panels operating at 79 F produce a detectable increase in the temperature of the clothing surface of the lower extremities. Under these conditions the gradient between extremities and environment is increased about 28 percent over comparable heating with ceiling panels operating at 95 F.
2. Foot temperatures under comfortable conditions requiring no house heating are about 10 deg above shoe surface temperatures which are near 74 F when floors are at 71 F, other surfaces and air temperatures being within 3 deg of this value.
3. Physiological considerations are reported which support the view that floor temperatures above 75 F are not desirable.
4. Since ceiling location of the radiant panel does not produce a significant effect on head temperature, this location for a radiant panel is preferred to the floor location.

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DISCUSSION

T. N. ADLAM, New York (WRITTEN): I can concur most heartily with practically everything which is disclosed in this particular article, including most of the conclusions.

For many years, I have claimed that high floor temperatures are detrimental to the vitality of the occupants, and during the last two or three years I have received numerous letters complaining about hot floors, both in residences and industrial plants. It is correct that in England a floor surface temperature of 72 F has been considered the maximum, although I have always held the opinion that this was somewhat low for a maximum, and I do not know of any case where any ill effect was noted if the temperature did go slightly above 72 F under extreme weather conditions. There

is, I believe, one recorded instance in England where some members of an office building were known as tenderfeet, because it was necessary to maintain a portion of the floor at too high a temperature, and I had one personal experience in England of a school where steam pipes in the floor produced a number of complaints. However, after the heating medium had been changed to hot water, everything was satisfactory and no more complaints were forthcoming.

Several years ago, after considerable investigation, I came to the conclusion that in America 85 F was the maximum temperature which should be allowed for the surface of a heated floor, and in coming to this conclusion, I took into consideration that the average outside temperature throughout the winter would enable comfort conditions to be provided with an average floor temperature of 77 to 78 F. While this may be a little above the maximum temperature suggested by Dr. Herrington, I think it is sufficiently close to show a good relation between temperatures arrived at by making extended overall observations under actual living conditions, and the biological observations made by the authors of the paper presented.

It is correct that some people are more affected than others by warm floors, but my conclusions have been based on fairly large installations where the effect on a number of employees has been carefully noted. I note in the test records under the heading, *Black Body Temperature Room Center,* as recorded by the Globe thermometer, it gave an average of approximately 75 F. While this temperature may be satisfactory for the light clothing worn by the two male subjects, I feel sufficiently curious to ask, why this comparatively high temperature was chosen, because I find with a large office of female workers, such a high temperature would be considered uncomfortably warm.

While I advocate moderate floor temperatures only, I am not convinced that ceiling heat is always the ideal method of heating. In point of fact, from a heating engineer's point of view who prefers to choose from all the methods at his disposal, I am not convinced that any one position for heated panels can prove to be the ideal method in all cases.

In our own block of offices, we have one fairly large office 85 ft x 50 ft, with windows on three sides, in which 14 girls are engaged in general office work. We have both ceiling and floor coils so that these can be used simultaneously or individually, and while the temperature of the ceiling coils can be raised to give sufficient heat from the ceiling alone to meet the total heat loss requirements from this office, we do get complaints from some of the girls in cold weather, that when the ceiling coils alone are in operation there are cold spots near the windows facing North and Northeast. These windows have double glazing, and the room as recorded by a Globe thermometer, can be maintained at a temperature of 70 to 72 F without difficulty. Under similar conditions, with the same room air temperature and the floor coils also in operation, but with a lower ceiling temperature, there are no complaints whatever. In my opinion, the most ideal conditions of comfort seem to exist in any room when the floor coils are operated to give a moderate surface temperature, and the additional heat is coming from the ceiling or walls.

Speaking strictly as a heating engineer, I think every installation should be considered on its merits. I do not think that true comfort can be obtained unless the floor is sufficiently warm and protected from drafts to prevent cold feet, because even though the floor may be heated and thick rugs are placed on the floor, a small movement of air passing over the floor, possibly created by descending currents of cool air from windows or other cold surfaces, will be very objectionable. Of course, it is not always necessary to have heated floors for true comfort. Providing the floor surface is sufficiently warm and there are no currents of air over the floor surface, and no considerable conduction of heat from the feet to the floor, all will be well.

No doubt, Dr. Herrington will tell us that cold feet will not be as detrimental as a very hot floor, a statement with which I would readily agree. Nevertheless, cold feet do destroy the real comfort which we all wish to enjoy. I think, therefore, that

every heating engineer should consider what type of heating will best suit the building and its occupants, whether it be radiant panels in the floor, walls or ceiling, or whether it be a combination of radiant heat and convected heat.

Finally, to all heating engineers and architects, I do commend Dr. Herrington's conclusions on the ill effect of high floor temperatures, because it does undoubtedly retard the agility of work people and exposes the occupants to a greater shock when going from a room with a hot floor to the cold and wet surfaces of the streets and sidewalks in winter.

J. S. LOCKE, Minneapolis, Minn. (WRITTEN): If the conclusions presented in this paper prove to be completely valid it would appear that physiological effects and practical problems of construction would indicate that floor panel heating should not be used in any except the mildest climates. Needless to say, this would require a considerable change in present design practice.

There are one or two comments and questions which I would like to bring out.

1. Conclusion 3 of this paper is the one which seems to have the greatest effect on panel heating design. *Physiological considerations are reported which support the view that floor temperatures above 75 F are not desirable.*

This appears to be based on the reduction in vascular tone in the lower extremities which may be caused by higher temperatures—which in turn reduce the blood supply for other parts of the body. It is my understanding that different conditions of vascular tone usually exist in different parts of the body at any given time. If this is so, might it not be possible that the total amount of blood supply required for thermal balance in the body would be approximately the same regardless of which body area is subjected to local heating so that an adequate supply would still be available for other bodily functions.

2. In Table 1, the black body temperature is indicated as about 0.7 F above the ambient temperature and also 0.7 F above the average of floor and ceiling temperatures. If the walls are below room ambient temperature (as is indicated in the paper) it would seem that the black body temperature should be no greater than the average of floor and ceiling temperatures unless there is considerable effect on the globe thermometer from the room occupants. Further clarification would be appreciated.

3. Since conclusion 3 is so different from accepted design practice it would seem that considerably more data should be collected.

R. G. VANDERWEIL, Boston, Mass. (WRITTEN): It seems greatly encouraging, after many years of discussions of the problem *ceiling vs. floor panels* to find actual test data analyzing the physiological conditions prevailing with both heating systems. The authors should be commended, particularly since their results are not only given on a *comfort scale,* but also translated to the less ambiguous terms of body surface temperatures and heat losses of lower and higher extremities.

For many years I have expressed the opinion that from the comfort point of view, ceiling systems are superior and that where floor systems must be used for budgetary reasons, their design temperature should be maintained at 85 F or below. In fact, in my experience during the past two years, there were several occasions where floor systems had to be used and I found little difficulty in convincing the owners and architects to provide additional insulation or double glass in order to maintain floor temperatures below this level. This paper clearly indicates that floor temperatures should be maintained at a level lower than 80 F (and not 85 F). However, this would call for extensive insulation of buildings located in the central and northern United States. Then again, I believe that an assumption of 85 F floor temperature for design outside temperatures should still be tenable, in consideration of the fact that these temperatures occur only several days per heating season. Hence, it can be assumed that the heating system, if properly controlled, will operate on floor temperatures considerably lower than design temperature.

One of the authors' findings was read with great interest, namely, that with virtually *equal* temperature conditions, the sensation of the occupants was on the warm side but *cooler* with ceiling heating. At this point I should like to add that similar tests performed at air and black body temperatures below 70 F (*i.e.* around 67 F rather than 74 F as tested) if made in addition to those reported, would give additional enlightenment on the subject, *floor vs. ceiling panels.* Although my predictions in the past have not proved *76.3 correct,* I do believe that the ceiling system will be considered *less cool* by the occupants at such conditions, just as it was considered *less hot* at the conditions tested.

The following figures will confirm statements made by others: As the black body and air temperatures of Tables 1 and 2 indicate, the radiation component of panel heat transfer is considerably greater with ceiling panels. More important than this, I feel, is the comparison of increase of head and lower extremities temperatures of same tables, if we consider that the increase of the surface temperatures of the lower extremities with floor panels is exactly twice as great as the increase of head surface temperatures with ceiling panels.

If we further consider that natural sun radiation under out of doors conditions must cause a similar increase of head surfaces temperatures, it may well be restated that the ceiling system seems to approximate closer outdoor comfort conditions than its floor counterpart.

Incidentally, I would like to point out the fact that floor temperatures in the test room were maintained at 78 F with an *outside climate* temperature of 35 F and an inside temperature of approximately 74 F. The small differential between panel and room temperature (4.5 deg with floor, and 11 deg with ceiling heating) seems to indicate a space slightly better insulated than the average building. Is this the case? Furthermore, the gradient ceiling panel-room temperature was 2.5 times greater than the gradient floor panel room temperature. Is this due to use of warmer wall temperatures in case of the floor panel?

One of the most interesting findings of the authors was made in connection with the increase of temperature gradients between respective head and lower extremities *vs.* environment temperature. It appears that with floor heating, the heat loss of the lower extremities increases by 18 percent; with ceiling heating the losses of the skull by 5 percent. Could this be the basis of occasional complaints in connection with floor temperatures exceeding 85 F?

It is possible that ceiling panels would be stimulating where activity and alertness is required, such as in office buildings, etc. and that floor panels of moderate temperature might be favorable where the occupant wants to feel at ease. If this statement is true, it should prove to be additional incentive to construct two-story homes in such a manner that one single coil located in the first floor ceiling could be used to heat both upper and lower floor. Few cases are known where this method is used, and still, by application of concrete floors and proper selection of flooring, conductance and insulation, a single coil could provide ceiling heating for the living spaces and floor heating for the bedrooms.

C. A. HAWK, JR., Pittsburgh, Pa. (WRITTEN): In my opinion, the conclusions drawn from this test are incorrect and are invalidated by a fundamental error in the conception and planning of the program.

The selection of an arbitrary temperature (at the center of a blackened globe thermometer located at the room center) as the *primary condition of comparability* around which the whole study revolves, produces an artificial set of conditions radically different from those prevailing in comfort heating systems. The study thus becomes one of academic interest only, without even directional significance for those interested in the practical problems of space heating.

The principle of an arbitrary thermostat setting (*e.g.*, 75 F—no more, no less) has been opposed by us for many years; we have advocated that the room occupant

set the control for the most comfortable temperature and change it if he becomes too warm or too cold.

In the test program under discussion, the result of this faulty conception is clearly born out by the mean values of the *Mean Comfort Vote* and *Mean Temperature Vote* columns. The results for the ceiling system tests seem to place that series at about the right level of comfort but, in the floor series, the room was too warm for comfortable occupancy. Would not a much more useful test have occurred if the researchers had chosen a statistically even comfort series?

The conclusions based on these data seem quite unwarranted on the basis of such meager and ill-planned laboratory study, particularly since the conclusions are counter to the overwhelming experience in this field. After several years experience in this work, I have yet to encounter the slightest evidence of discomfort or physical deterioration with properly designed and controlled floor-type radiant heating systems.

P. S. PARK, JR., Pittsburgh, Pa. (WRITTEN): I am in agreement with Mr. Hawk in his claim that the research work of Herrington and Lorenzi contains a basic error which directly affects the test results and conclusions. I believe the test and its results have taken two different things and judged them by a standard which is favorable to one, and the result is obvious *before the judging*.

I think it has been proved beyond doubt that a warm floor and a warm ceiling have different heating characteristics. In a given room having a fixed heating load, a floor type system will vary from a ceiling system in heat transfer by convection, heat transfer by radiation, emissivity, and surface temperatures. We should not therefore control the two by the same thermostat, set for the same reading, and located at exactly the same geometric spot, and expect to come to any useful conclusion except that the location and setting of a thermostat have an effect on the comfort conditions. In the test work by Herrington and Lorenzi the thermostat location and/or setting were apparently not ideal for the floor system and I can therefore see no reason why the floor system should have produced ideal comfort conditions under the handicap thus imposed.

I do not mean to infer that there is a great difference between the floor type and a ceiling type system. Actually, the difference need only be slight to compensate for the small percentage variations reported in the test results and the almost immeasurable difference between the abstract comfort classifications of *warm, comfortably warm,* and *comfortable.*

I should also like to question the authors' statement that *in floor heating a larger proportion of heat is being lost to the environment through the legs.* This statement is apparently based on the 1.78 deg higher mean temperature of the lower extremity and relatively constant mean air temperature at the 36 in. level. There appears to be no consideration for the fact that the temperature of the floor—to which heat is lost by radiation—is an average of 5.74 deg higher with the floor system. Neither is any mention made of the higher temperature of the air—to which heat is lost by convection—around the feet and ankles.

In conclusion, I want to state that my greatest concern over the published conclusions reached by the authors of the test, is the suggestion that floor surface temperatures over 75 F may be physically harmful. I am neither a scientist nor a doctor, but as an engineer I failed to find evidence in the report which supported such a breath-taking decision. Consequently, I sought expert scientific opinion at one of the outstanding medical schools in Eastern United States. The report which I received just a week ago reads as follows:

The authors present no data to support their conclusion that the body cannot adjust to the temperature gradients which exist on skin surfaces exposed to heat of the maximum degree mentioned in the paper or that such gradients, if present, are harmful.

In other words, it is their opinion that the conclusions reached in the paper are not based upon data presented therein. Those members of our faculty who reviewed the paper feel that such a point might be substantiated by a properly planned experiment station, but they are unanimous in believing that the present paper does not give adequate experimental support to the conclusion reached.

LESTER T. AVERY, Cleveland, Ohio: I think the important point here is not so much the argument as between *floor vs. ceiling panels,* but the point that when you have radiant heat or panel heat in the floor or in the ceiling, the room temperature for comfort is very close to the panel temperature.

The original conception that I think Dr. Winslow gave us from some tests many years ago indicated that with warm panels you could carry cold rooms and have equal comfort or great comfort. In fact, there were news articles published, purporting to show that you could keep a room in a home at 60 F and be more comfortable than at 80 F because the floor itself did the heating.

This paper points up that these temperatures tend to equalize; and unless you have some miracle, they will always tend to equalize.

L. N. ROBERSON, Seattle, Wash.: I have lived in a radiant heated home for the last ten years. We more or less backed into this radiant heating and so, when we put it in, arranged our living room so that we could turn on a quarter, a half, or all the heat in the ceiling panel, and the same with the floor. This is electric radiant heat, incidentally. We found that the ideal combination—by the instruments—was one-quarter in the floor and one-half of the total heat in the ceiling. In other words, about three-quarters of the normal design as you would calculate it.

My first victim of improper panel design was my own wife who is a decided blonde. We found the ceiling panel was as critical as the floor, and that it is necessary to hold the panel temperature down below 85 F for the light complexioned and the folks with the high foreheads. So I think we should give some consideration to the panel temperature from a psychological standpoint on both the floor and the ceiling.

In our office we have set a maximum panel temperature for the extreme design condition of 85 F. In most areas this occurs for only a very few days during the year, and the normal temperature on the panel is somewhere in the range of 70 F to 75 F, which would agree somewhat with the findings of these gentlemen.

One other thing we did find was that high panel temperatures where you have a carpet pad under the carpet produced a very undesirable odor. I do not know just how to explain it. It is just a very bothersome carpety smell that occurred when the temperature would go much over 75 F.

G. LORNE WIGGS, Montreal, Canada: We have been doing radiant heating in Canada for six or eight years, starting with the floor type installations and then changing to installations in the ceiling.

We find one thing that was not brought out in this paper, but I think some attention should be paid to it, in considering a floor heating installation. We have found that where the factory workers were walking up and down the floors, such as the girl at the textile machine, that she could quite easily and comfortably stand temperatures up to about 85 F; but when that same girl was raised to be a supervisor and sat down at a desk on the same floor, most of the time she very quickly complained of the excessive heat of the floor. Something that should be very carefully noted, I think, in all these tests, is whether the occupants are sitting still or whether they are moving about.

We have many installations, or numerous buildings, I should say, in which some floors have the radiant heating in the floor, and others have it in the ceiling. From our experience and judging mostly from what complaints we have received, the ceiling job has invariably given a better or more comfortable condition.

We do not run air temperatures in Canada anything like you do here. 75 F in Winter might kill most Canadians. The temperatures which we probably maintain on most of our radiant heating systems, that is, the air temperatures, are of the order of about 67 or 68 F. To get that we run ceiling panel temperatures on the average of about 90 F with a maximum of 110 F in extreme sub-zero weather; and by testing many installations with the double windows that are used there and the general construction adopted in Canada, we find that the floor temperatures are consistently maintained at about 72 F and that gives great comfort.

I disagree with Mr. Adlam and the other speakers who mentioned that 85 F was or could be comfortable under all conditions; and I know that it causes not only discomfort, but also excessive perspiration of the feet. We have even heard people complain of the deterioration of their shoes from that perspiration. Medically, I have not checked up on it.

I believe that the original patents on radiant heating were taken out in England in 1909; and the English have had a great deal of experience since that time. As head of a firm of consulting engineers in Canada, we brought from England two engineers who had been designing radiant heating systems there for many years; and from these men we have learned a lot about English radiant heating practice.

In England the thermostats do not have thermometers on them, and there is no calibration on the bottom to indicate what temperature the thermostat might be set for. One end says *warmer* and the other *colder*, and it may be set for the comfort zone desired. It is not a question of degrees.

To summarize: For people standing or sitting in a steady state, I am in agreement with the authors that some temperature around 75 F is about the maximum; and only provided the occupants are moving around enough, temperatures higher than that could be permitted.

AUTHORS' CLOSURE: We appreciate the discussions that have been given to this paper. It is in no way an attack on floor heating, nor was it influenced by any prejudice in favor of ceiling heating.

The temperature of 75 F selected as a mean comparison condition is a moderate one for resting subjects without coats, and seated in chairs providing minimal insulation. Such chairs were used because it is quite simple to define an uninsulated chair, and very difficult to choose an upholstered example representing average heat insulation. Coats were removed for a similar reason. Men's suits vary considerably in insulative value, and the coatless situation affords a more reproducible test situation. Actually, the heat loss of such subjects at 75 F is quite comparable to that of subjects with coats at 72 F.

The subjects of the experiment were young men; and they were the same young men under those conditions in which the total heat balance between the body and its environment was in all cases identical and reasonably satisfactory. They showed these differences as the result of the differential effect on the different parts of the body.

The wall temperatures were not, of course, very low. The walls did, as it has been suggested, have insulative value; and the wall temperatures were not extremely cold. It did not make any difference because those conditions were identical throughout the experiment. The only thing significant is the difference in the direct effect; and while these people were all having essentially the same thermal interchange with the environment as the total, they had to accomplish it in the one case by increasing considerably the circulation to the feet as a result of this overheating effect on the feet.

There was no difference in the overall heat loss at all between the two groups of subjects. The point is that with two situations of similar overall heat exchange, the 79 F floor temperature results in a 28 percent increase in the amount of heat loss by the lower extremities whereas the ceiling panel only caused 5 percent increase in the heat loss from the head and face. Those are the fundamental facts. It is not a matter of *total* heat loss, but of the *differential* heat loss between the different

regions of the body. The total temperature interchange was the same in the two cases. In one case, the body, in order to meet this overheating in the lower extremities, had to change the distribution of the blood supply substantially which is a distinctly undesirable condition.

For the benefit of engineering interests, I would like to point out that many of the good points of floor heating, *i.e.*, low vertical gradients and comfort range floor temperatures in basementless houses can be had in systems with moderate floor temperatures. Such designs use a warm air traverse beneath the floor, or in addition to floor coils, introduce a portion of the required heat through accessory radiators, or as a third instance, concentrate heat in floor regions near the exposed walls where foot contact is not common.

Since physiological heat must be lost from all parts of the body, it must be obvious that at contact temperatures of 85 F, the foot must be substantially above this value. Foot temperatures of the order of 90 F are common in summer acclimatized individuals, but are neither typical nor desirable for occupants under conditions of winter heating.

It is obvious if you have a floor system in a home operated between 70 and 75 F with a cold day floor temperature of 85 F, that such a temperature is not harmful; but the evidence is good that 85 F is too high a general design temperature. I think these results show that the temperature of the floor part of the panel system should not be over 75 F or at the most 80 F.

There is nothing in these experiments that should be construed as an argument for ceiling heating as opposed to floor heating. The results merely show that you can heat in a great many different ways.

Actually, our offices in the new wing of the Pierce Laboratory are heated by wall panel heat. You can use any combination, but a combination ought to be adapted to the particular design. The principal point is in estimating the fraction of the total panel heating load that should be carried by the floor; this should not require that the temperature of the floor be over 80 F, preferably not over 75 F, except for unusual conditions.

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No. 1391

RESISTANCE GRADIENTS THROUGH VISCOUS COATED AIR FILTERS

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This paper is the result of research sponsored by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the Graduate School of the University of Minnesota in cooperation with the Engineering Experiment Station, University of Minnesota

THE performance characteristics of air filters are markedly affected by the type of dust to which the filter is subjected and by the manner in which this dust is distributed throughout the filter. The majority of unit filters of either the viscous or dry type are constructed of a series of layers of filtering medium through which the air-dust mixture must pass. During this process the air entering the filter is split into a series of minute streams of tortuous passage. By virtue of the continually changing direction of these air streams, the heavier dust particles are literally centrifuged from the air onto the surrounding medium.

If the material from which the filter is constructed is closely woven and possesses only extremely fine interstices, most of the dust is eliminated at the initial surfaces, and it then becomes necessary to design a filter with a comparatively thin, usually dry, medium, but with a greatly increased inlet surface area for collection of the dust. This is usually done by *accordion pleating* the medium. The distribution of dust is primarily across the initial surfaces with the resistance drop across the filter kept low by virtue of the large surface area. With such filters, there is evidence to support the theory that the process of dust elimination is still brought about through impingement and centrifuging action, although it appears probable that some of the dust is removed by an actual screening process in the passage of the air-dust mixture through the extremely fine openings.

Viscous or dry filters constructed of coarsely woven layers of material, usually coated with a viscous adhesive, vary in thickness from one to four inches. An

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Presented at the 56th Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Dallas, Texas, January, 1950.

attempt is frequently made to grade these layers from coarse to fine, passing from the entering to the leaving surfaces. In some cases, a grading of adhesive has also been attempted with the heaviest coating at the discharge side of the filter. This gradation of fiber size, density, and coating is adopted in an attempt to distribute the dust collected more evenly throughout the filter. The coarse, more thinly oiled entering medium of the filter is the least efficient in removing dust and will therefore collect only the heavier and larger dust particles. The leaving surfaces of denser pack and heavier oiling will be more efficient in collecting the finer, smaller dust particles which are more difficult to remove from the air.

Theoretically this gradation of efficiency should result in a corresponding gradation of dust particle size collected throughout the filter. When properly

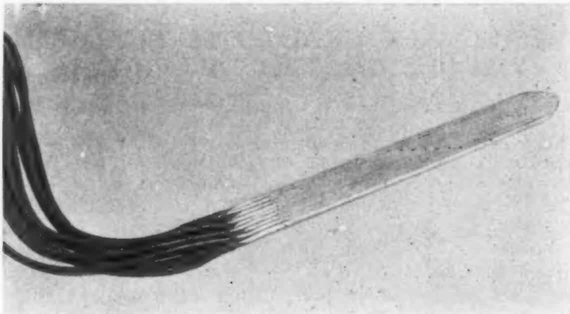

FIG. 1. RESISTANCE GRADIENT MEASURING DEVICE

designed for a specific dust, a filter should possess maximum efficiency, coupled with long life and dust holding capacity. In contrast, a filter constructed without gradation in medium, oiling, or both, will have the same dust removing efficiency at all layers of the medium, and the dust will tend to collect in the initial layers and result in a high, localized resistance at such a section of the filter. It is thus possible to have most of the dust deposited on only a few layers with the rest of the filter practically unused at the time the final limiting resistance drop across the filter is reached.

MEASUREMENT OF FILTER RESISTANCE GRADIENTS

The dust loading of filters at the various sections of the filtering medium is usually determined by examination. However, such a procedure is at best capable of only approximate, and in some cases even faulty conclusions. No true quantitative picture can be obtained by such means.

In order to obtain measurements of the actual resistance gradients through unit filters under various conditions of loading, a special measuring device was constructed for use in these tests. This device (Figs. 1 and 2) consists of nine ⅛-in. copper tubes laid parallel to each other with the space between the tubes

filled with solder to provide flat surfaces on either side of the pack. These tubes are sealed at one end, with the sealed ends of the tubes arranged to form a point to facilitate insertion into the filter.

Each of the measuring tubes constituting the pack contains a 1/16-in. hole drilled through the side to make possible the measurement of the static pressure at various points within the filtering medium. The first and second holes are spaced ¾ in. apart in the direction of the air flow. The next six static openings

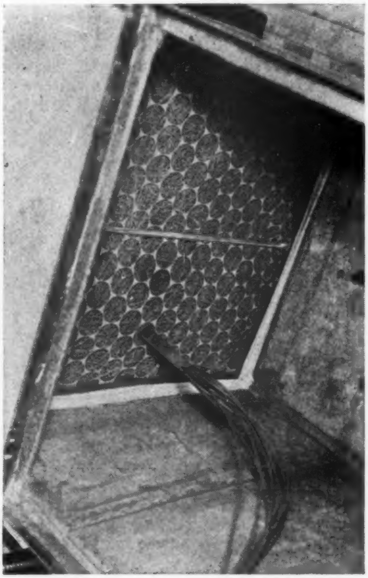

FIG. 2. DEVICE POSITIONED IN FILTER

are spaced ¼ in. apart; and the last opening is located ¾ in. farther along in the direction of air flow. This spacing was arranged for the measurement of the resistance gradient through a standard 2-in filter. This makes possible the measurement of the static pressure ½ in. upstream from the entering face of the filter, ½ in. downstream from the leaving face of the filter, and at each ¼ in. through the filtering medium itself.

The open end of each of the measuring tubes is connected by means of a flexible rubber tube to an inclined draft gage for measurement of the resistance gradient. All tubes are continued 2¼ in. beyond the last opening and 2¼ in. upstream from the first opening in order to introduce as little turbulence as possible at the points of measurements. The entire device is approximately 9⅜ in. long, 1⅛ in. wide, and ⅛ in. thick.

TEST APPARATUS AND PROCEDURE

In order to develop a procedure by which typical resistance gradients through unit filters might be determined under various laboratory test conditions, the weight method of testing air filters was adopted as a basic test. This test apparatus, shown in Fig. 3, has been described in detail together with the test procedure in a previous publication.[1] The apparatus consists essentially of a test duct, 20 in. square and 9 ft long, connected to an exhaust fan by means of a

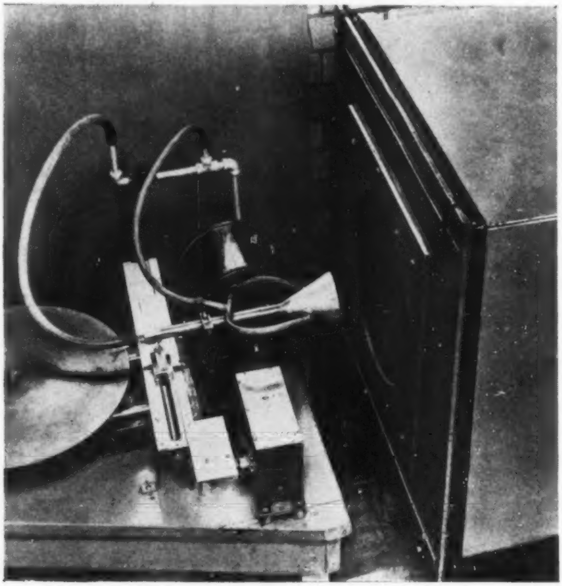

FIG. 3. VIEW OF AIR FILTER TEST APPARATUS
DUST FEEDER

reducing section 3 ft in length and a 12-in. diameter duct 5 ft in length. The test filter is sealed in the square duct approximately 8 in. from the reducing section. A blower is used to draw the air through the filter, and the volume of air is measured by a standard bell-shaped orifice; the total pressure drop across the filter is measured by two piezometer rings connected to a standard inclined water gage.

The dust feeding apparatus shown at the left of Fig. 3 consists of a revolving disk upon which the dust sample has been evenly distributed in a layer of uni-

[1] A.S.H.V.E. RESEARCH REPORT No. 1094—Air Filter Performance as Affected by Kind of Dust, Rate of Dust Feed, and Air Velocity Through Filter, by Frank B. Rowley and Richard C. Jordan. (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938.)

form thickness. The disk containing the dust sample is placed on a rotating plate and a continuous, uniform ribbon of dust is shaved from the outside edge of the dust ring by a knife edge fastened to the end of an air suction tube. Air and dust are inducted through this tube by connecting it to a low pressure air jet and venturi tube. Throughout all the tests the filter face air velocity was maintained at 300 fpm and the dust rate of feed to the filter at either 20 or 40 g (grams) per hour depending upon the test. In all cases, standard determinations were made of the filtering efficiency, life in hours, and the dust holding capacity in grams, with the latter two determinations based upon a limiting resistance of 1.0 in. of water.

In addition to the standard test procedure for determinations of the efficiency, life, and dust holding capacity, resistance gradient measurements were made throughout the tests by the device previously described. It was found that great care had to be exercised in inserting the gradient tube in the filtering medium in order to obtain true values of the resistance gradients. In some cases it was found that the tube could be inserted satisfactorily by merely pushing it through the medium, while in others, it was necessary to sever the medium with a knife before insertion.

It should be emphasized that in any case where a tube, such as the one used in the present tests, is inserted in a medium through which the air is flowing, there are certain to be minor alterations in the nature of the flow. It is likely that at some points there will be at least a partial restriction of the static openings by the surrounding medium. However, it was found that if care was taken in positioning the gradient tube, a remarkably smooth gradient curve could be obtained. The total pressure gradient across the filter, as determined by means of the gradient tube, checked very closely with the total pressure drop as determined by means of two piezometer rings located upstream and downstream of the filter.

In several cases it was necessary to insert the gradient tube through the media at two or three different points before a satisfactory position was located. In all cases if the gradient curve as determined for the unloaded filter prior to starting the test was smooth and indicated no abnormalities, and if the total pressure drop as indicated by the gradient tube checked with the total pressure drop as indicated by the piezometer rings, it was assumed that the position of the tube was satisfactory.

Several tests were conducted in order to determine the effects of lint on filter pressure gradients. The procedure used in such tests was in accordance with that described in a previous paper,[2] and consisted essentially of injecting lint fibers into the test duct by means of a special feeder concomitantly with the feeding of the test dust. In these tests the lint was fed at a rate of 2 g per hour and the dust at a rate of 18 g per hour for a total of 20 g per hour.

Test Dusts

Since the purposes of the present tests were to develop a procedure for determining pressure gradients through filters and to indicate the effects of various

[2] A.S.H.V.E. Research Report No. 1145—The Effect of Lint on Air Filter Performance, by Frank B. Rowley and Richard C. Jordan. (A.S.H.V.E. Transactions, Vol. 46, 1940.)

test dusts and loadings upon the resistance gradients and filter performance characteristics, several different artificially synthesized dusts were used. One of these consisted of a mixture of 50 percent by weight of Pocahontas ash screened through a 200 mesh screen, 20 percent by weight of Illinois fly ash screened through a 200 mesh screen, 20 percent by weight of lampblack screened through a 100 mesh screen, and 10 percent by weight of Fuller's earth screened through

TABLE 1—FILTER PERFORMANCE TEST CHARACTERISTICS

TEST No.	TYPE OF FILTER	TEST DUST	RATE OF DUST FEED GRAMS/HR	AVG EFFICIENCY %	LIFE, HR[a]	DUST HOLDING CAPACITY[a] GRAMS
1	A	50% Pocahontas ash 20% Illinois fly ash 20% Lampblack 10% Fuller's earth	40	89.7	10.0	358.8
2	A	100% Carbon black	20	73.1	32.4	473.8
3	A	80% Pocahontas ash 20% Carbon black	40	90.0	17.0	611.9
4	A	90% Dust 10% Lint Dust {80% Pocahontas ash 20% Carbon black}	Dust: 18 Lint: 2	92.6	2.6	48.3
5	B	90% Dust 10% Lint Dust {50% Pocahontas ash 20% Illinois fly ash 20% Lampblack 10% Fuller's earth}	Dust: 18 Lint: 2	91.5	1.7	31.2
6	C	80% Pocahontas ash 20% Carbon black	40	79.4	21.3	706.8
7	C	90% Dust 10% Lint Dust {80% Pocahontas ash 20% Carbon black}	Dust: 18 Lint: 2	83.4	4.5	75.1
8	D	Carbon black	40	32.2	66.0[b]	850.8[b]
9	E	90% Dust 10% Lint Dust {80% Pocahontas ash 20% Carbon black}	Dust: 18 Lint: 2	89.4	3.1	55.5

[a] Life and dust holding capacity based on limiting filter resistance of 1.00 in. of water.
[b] Life and dust holding capacity based on limiting filter resistance of 0.87 in. of water.

a 200 mesh screen. This mixture was one of the earliest used in filter testing at the Experimental Engineering Laboratories at the University of Minnesota and was introduced into these tests merely to increase the range of test dusts.

Another mixture used and developed more recently in the University of Minnesota laboratories consisted of 80 percent Pocahontas ash screened through 200 mesh and 20 percent carbon dust[3] screened through 100 mesh. In addition to these two mixtures some tests were run using the carbon black without mix-

[3] A.S.H.V.E. RESEARCH REPORT No. 1218—Overloading of Viscous Air Filters During Accelerated Tests, by Frank B. Rowley and Richard C. Jordan. (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942.)

ture; other tests used either the 50-20-20-10 or the 80-20 mixture in conjunction with lint. The lint used consisted of Java kapok fibers cut into lengths ranging up to 3/16 in. These fibers, as indicated in earlier publications, reproduced reasonably well, both as to length and to diameter, actual lint fibers removed from commercial and residential filtering installations.

The densities of these test dusts were obtained by tapping containers holding the dust to maximum density and were as follows: Pocahontas ash (200 mesh), 0.77 per cc (cubic centimeter); Illinois fly ash (200 mesh), 0.74 g per cc; lampblack (100 mesh), 0.16 g per cc; Fuller's earth (100 mesh), 0.60 g per cc; K-1 carbon black (100 mesh), 0.48 g per cc.

TEST FILTERS

A comprehensive study of the test dusts on a complete range of the different types of filters available was not attempted, nor were all of the filters chosen tested with each type of dust. Selection of the filters, of the dusts, and of the tests actually conducted was made only to obtain a range of combinations broad enough to indicate the value of this type of analysis and to indicate typical results which can be expected. Five filters were involved in various tests:

Filter A—a viscous coated, throw-away type filter 2 in. thick. The straight, fibrous medium used in its construction was graded in fiber size, density and oiling from entering to leaving side.

Filter B—a modification of filter A with a curly instead of straight, fibrous medium used in construction.

Filter C—a cellular type filter, 2 in. thick, built in two sections each with the angle of the cells set at 45 deg to the centerline of the duct and 90 deg to each other. The cells on the entering side were of larger dimensions than those on the leaving side.

Filter D—a 2 in. thick, throw-away type filter with a light, viscous coating. The filtering medium was wood fiber graded in density but not in size or oiling from entering to leaving side.

Filter E—a 2 in. thick, expanded paper, throw-away filter graded in oiling from entering to leaving side.

Some of these filters were not standard types readily available on the market. All contained viscous coatings and all were sufficiently thick to permit measurements of the gradient curves. All contained a medium which was readily severed to permit insertion of the gradient measuring device.

TEST RESULTS

Typical tests results are presented but with no attempt to provide a comprehensive picture of the effects of any one test dust or the performance of any one type of filter. A tabulation of the conditions under which each test was performed is shown in Table 1 together with the average efficiency, the life in hours, and the dust holding capacity in grams. Data on filter life for the various tests are not directly comparable since the test conditions were prescribed by the best operating conditions to obtain resistance gradient curves.

In some cases the rate of dust feed was 20 g per hour while in others it was 40 g per hour. Doubling the rate of dust feed obviously had the effect of approximately cutting in half the life, in hours, of the laboratory test. The efficiency of the filter for 1 hour of test is the percentage, by weight, of the test

dust fed to the filter and eliminated by the filter. The average efficiency is the average of the hourly efficiencies determined throughout the test. The life of the filter is defined as the number of hours of test before an arbitrary limiting pressure drop across the filter, in this case 1 in. of water, is reached. The dust

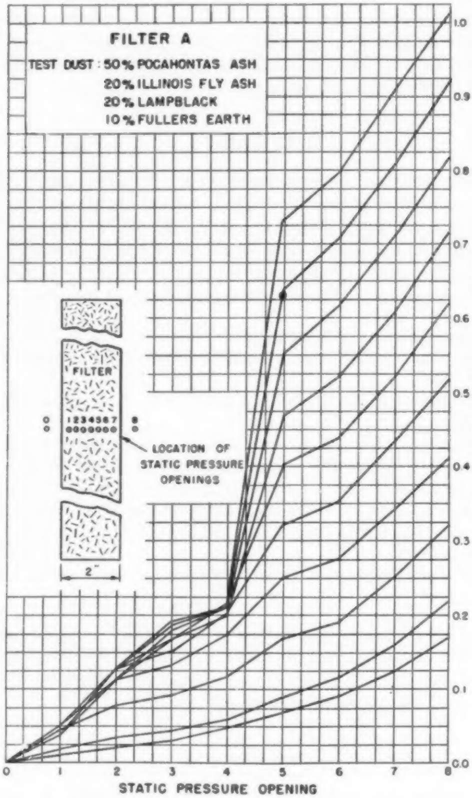

Fig. 4. Resistance Gradient Curves for Filter A

(Test dust: 50-20-10 mixture)

holding capacity is the number of grams of dust collected and deposited in the filter throughout the test life.

No attempt is made in this paper to discuss the effects of different test dusts, rates of dust feed, and other variables upon the performance characteristics of the filters, since this is covered in the literature[1, 2, 3]. Briefly, however, it may be

stated that earlier tests have shown that lint fibers are easily removed by air filters and are usually deposited in the entering layers of filtering medium. Carbonaceous dusts, particularly soot, as simulated by lampblack and to a lesser degree by carbon dust, are much more difficult to eliminate and are usually de-

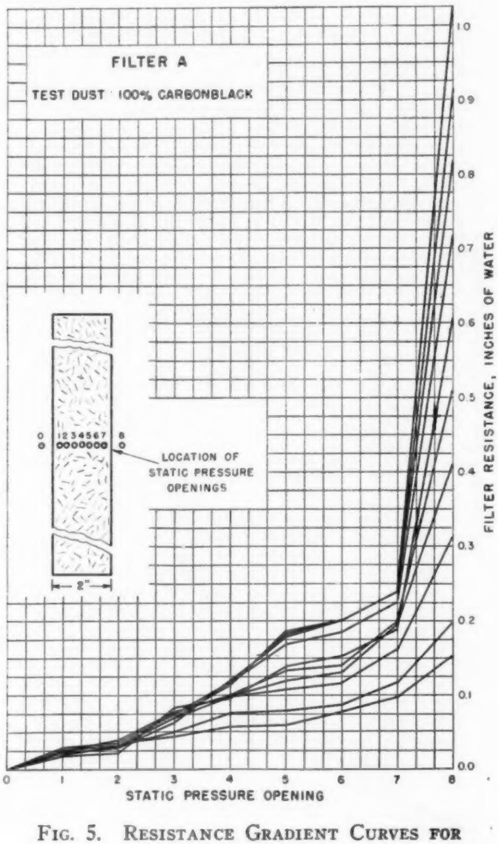

FIG. 5. RESISTANCE GRADIENT CURVES FOR FILTER A

(Test dust: carbon black)

posited heavily on the leaving surfaces of the filter pack where the filtering efficiency of the medium is higher. In most cases carbonaceous dusts are difficult to remove and the effect of their use is to lower markedly the overall efficiency of the filter.

Ash dusts of the type used throughout these tests are of intermediate size and

difficulty of elimination and are therefore found more generally deposited throughout the filter pack. The predominant factor involved in the difficulty or ease of elimination is the particle size itself. The heavier and larger particles,

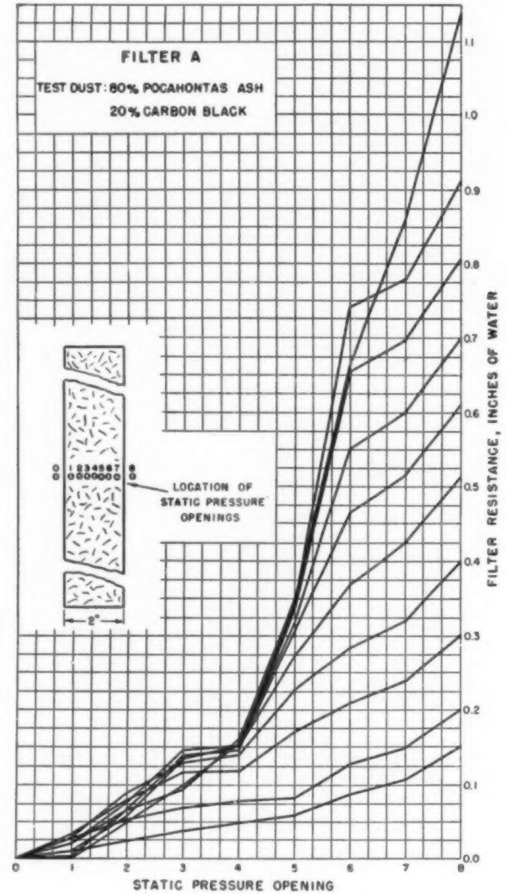

FIG. 6. RESISTANCE GRADIENT CURVES FOR FILTER A

(Test dust: 80-20 mixture)

such as lint or large dust particles, are more easily centrifuged and impacted against the oiled surfaces, whereas the finely divided carbonaceous dusts of small particle size possess a lower momentum and follow the tortuous passages

through the filter more easily. Furthermore, the larger particles with greater mass have a much greater effect per particle on determinations of efficiency based on weight than do small particles. A 10 micron diameter particle, for example, is much easier to eliminate than a 1 μ (micron) diameter particle, yet the

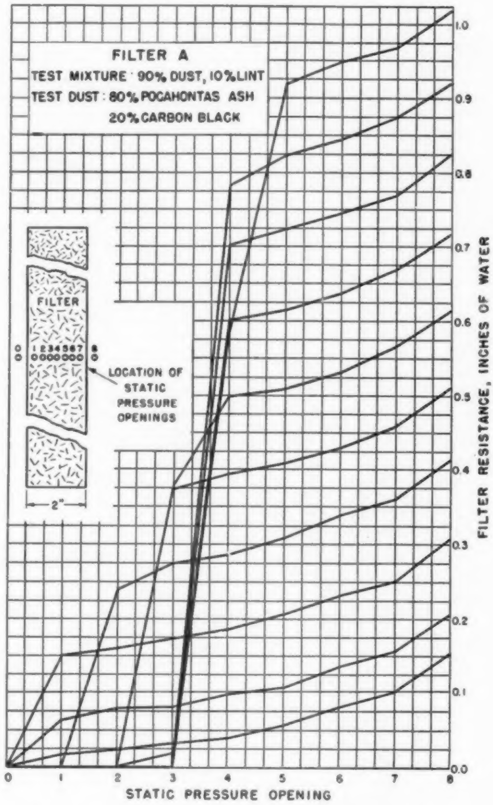

FIG. 7. RESISTANCE GRADIENT CURVES FOR FILTER A

(Test dust: 80-20 dust and lint mixture)

larger particle has a weight equivalent to approximately 1000 of the smaller units.

Figs. 4 through 12 present the filter resistance gradient curves for all tests reported. Positions 0 through 8 plotted as the abscissa on these curves represent the nine positions of the static pressure openings on the resistance gradient

measuring device shown in Fig. 1. The actual position of each of these static openings with relation to the filter is shown on the insert in each of these drawings.

Positions 0 and 8 were external to the filter, and the pressure drop between these two positions therefore represents the total filter pressure drop. The re-

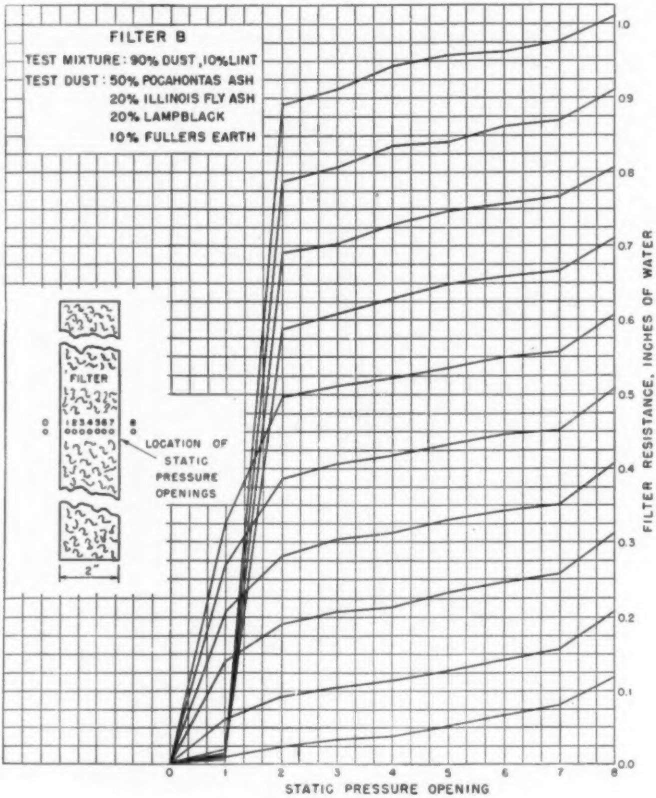

FIG. 8. RESISTANCE GRADIENT CURVES FOR FILTER B
(Test dust: 50-20-20-10 dust and lint mixture)

sistance, as shown at each of the intermediate points from 1 through 7, indicates the pressure drop to that point within the filter. The curve with the lowest total pressure drop on each figure indicates the initial resistance gradient through the filter before the test had started. The curve showing the greatest pressure drop represents the final resistance gradient through the filter at the end of the

test or at approximately 1 in. of water total resistance. Each of the intermediate curves represents the pressure gradients through the filter in total resistance incrⱥments of approximately 0.1 in. of water. Since it was difficult to stop the test at the exact 0.1 in. increment, the intersections of these curves with the

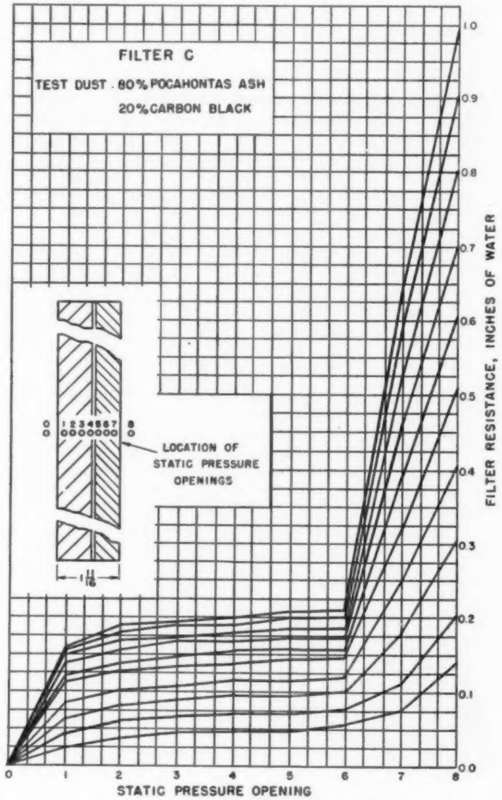

Fig. 9. Resistance Gradient Curves for Filter C

(Test dust: 80-20 mixture)

scale at position 8 do not coincide exactly with the scale increments. The slope of each pressure gradient curve between any two positions on the abscissa indicates the filter resistance rate of increase within that increment of distance. Thus, if the slope is low, there is little resistance in that section of the filter, whereas if the slope is high, the resistance is correspondingly great.

For filters A, B, and C, the initial resistance gradient curves indicate a higher slope at the latter layers of the filtering medium. This would be accounted for by any attempt to make the leaving layers more efficient by providing a denser pack or oiling. The initial resistance gradient curve for filter D indicates a

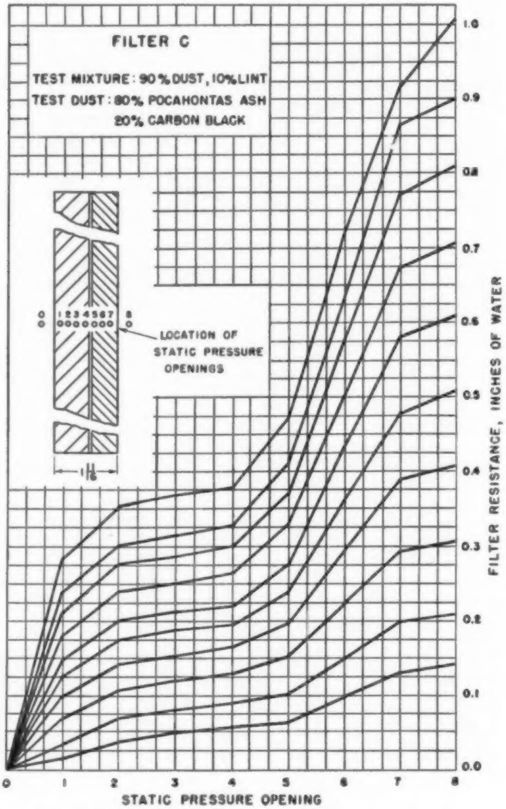

FIG. 10. RESISTANCE GRADIENT CURVES FOR FILTER C

(Test dust: 80-20 dust and lint mixture)

reasonably constant pack throughout the depth of the medium except toward the latter layers where the slope decreased, probably indicating a somewhat lighter pack or oiling. The ineffectiveness of this section of the pack is further borne out by the lack of any appreciable dust loading or increase in resistance in this section as the test progressed. The initial resistance gradient curve for

filter E indicates a reasonably constant pack. In several, but not all, cases, a marked increase at some point on the resistance gradient curve for the unloaded filter indicated a section of the filter where excessive loading of the dust was likely to occur.

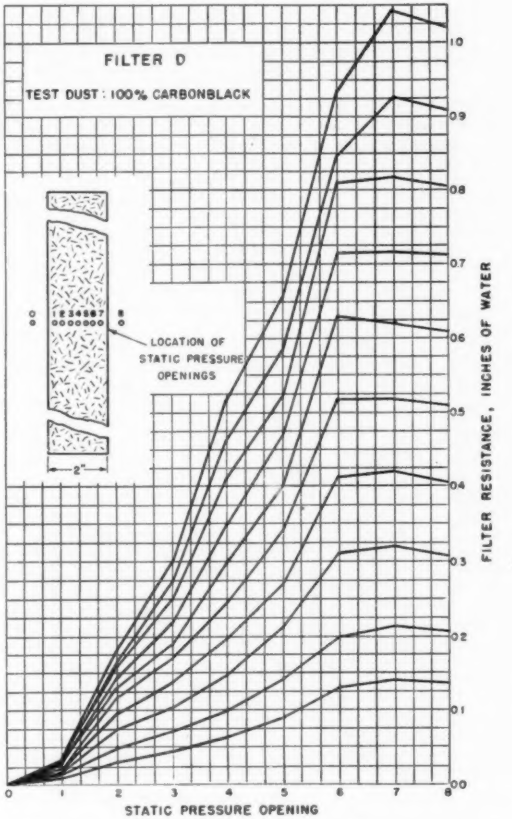

FIG. 11. RESISTANCE GRADIENT CURVES FOR FILTER D

(Test dust: carbon black)

Tests Nos. 1, 2, 3, and 4 (Figs. 4, 5, 6, and 7) were all performed on filter A with four different test dust mixtures. In test 1 the smoothly increasing initial resistance gradient curve gave no indication as to where the heaviest loading of the dust would occur by the end of the test. By the time the total pressure drop had reached 0.5 in. of water, the resistance gradient curve still indicated

a good distribution of the dust throughout the filter. However, from this point on, the dust loading in section 4-5 increased with a corresponding increase in the gradient slope. At the end of the test, over 50 percent of the total pressure drop occurred in section 4-5. Test No. 2 (Fig. 5) also on filter A, was made

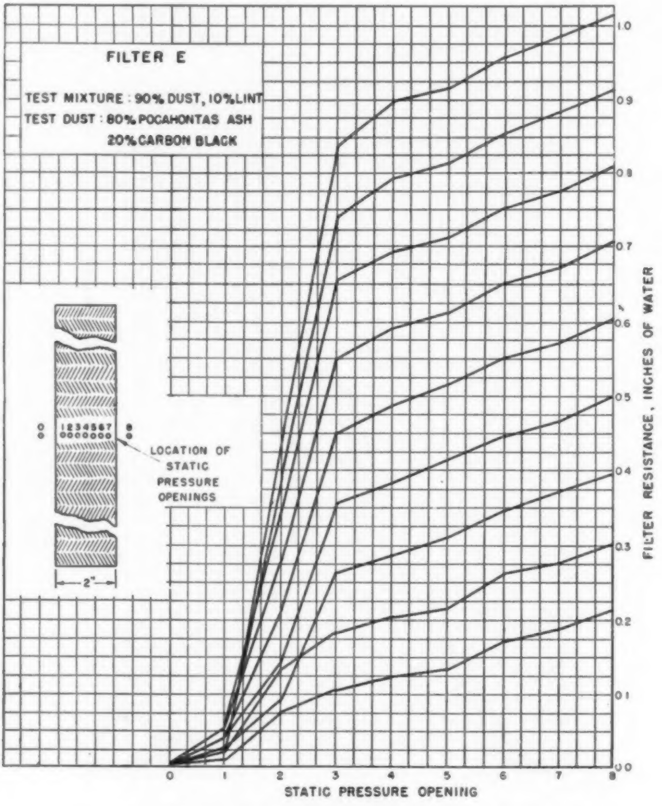

FIG. 12. RESISTANCE GRADIENT CURVES FOR FILTER E
(Test dust: 80-20 dust and lint mixture)

with 100 percent carbon black as the test dust. Since this dust is relatively difficult to remove from the air, it would be expected that the heaviest loading would occur in the latter sections of the filter. This proved to be true, and at the end of the test more than 70 percent of the total pressure drop occurred in the last quarter inch of the filter pack.

In Test 3 (Fig. 6) made on filter A with a mixture of 80 percent Pocahontas ash and 20 percent carbon black, the heaviest portion of the loading occurred

between sections 4 and 8. However, the resistance gradient curve at the end of the test indicated a more even loading of the dust than for any of the other tests on filter A. It would therefore be expected that the life and dust holding capacity performance characteristics for this test would be better than for any

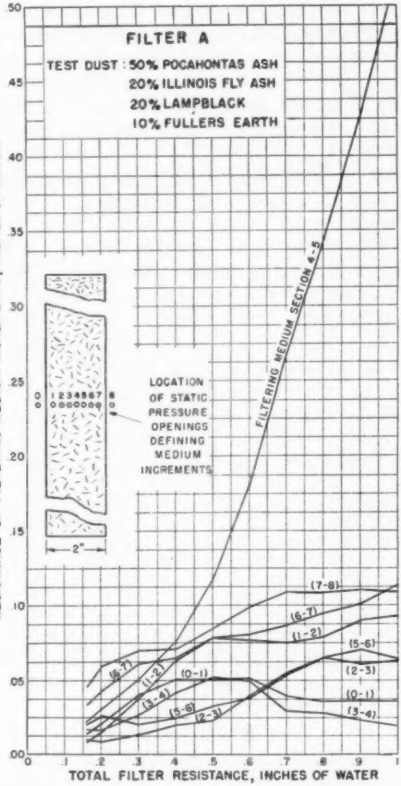

FIG. 13. RESISTANCE OF FILTERING MEDIUM INCREMENTS *vs.* TOTAL FILTER RESISTANCE (FILTER A)

(Test dust: 50-20-20-10 mixture)

of the other tests on this filter. This is borne out by reference to Table 1. Although the average efficiency is slightly lower than for test 4, it is not appreciably so, and the filter life (corrected for rate of dust feed) and the dust holding capacity were greater than for tests 1, 2 and 4 on the same filter.

Test 4 (Fig. 7) on filter A was performed with a lint-dust mixture. The filtering efficiency was somewhat higher than for the other tests on filter A since

the lint fibers formed a mat which, in turn, aided in filtering the dust. The heaviest loading occurred on the initial surfaces of the filter also as a result of this lint mat. Reference to Fig. 7 indicates a somewhat erratic resistance gradient condition as the test progressed, with a shifting of the sections of maximum resistance toward the interior of the filter.

At the end of test the greatest resistance gradient was reached between 3 and 5. This picture, however, is somewhat fictitious since a very marked compression of the filtering medium occurred during the test. It was noticed that in almost all tests as the filter loaded there was a slight compression of the medium so that the final thickness was somewhat less than the initial thickness. Normally, this compression was not more than 5 percent of the total initial thickness of the filter pack. However, in this test the initial heavy loading on the inlet surfaces of the filtering medium resulted in a total compression of 0.81 in. or 40.5 percent of the total thickness. The resistance gradient measuring tube was fixed in position relative to the leaving surfaces of the filter but was allowed to shift relative to the entering surfaces. Thus, the shifting gradient curves throughout the test indicate a continuing compression of the filtering medium rather than a shifting of the dust and lint load within the filter. At the end of the test the static opening at position 3 had actually emerged from the medium.

Test No. 5 (Fig. 8), performed on a modified form of filter A with a somewhat different filtering medium and again with 10 percent lint in the test mixture, also indicated some degree of compression of the pack during the test. In this case again, most of the loading occurred at the initial surfaces of the filter, and the degree of compression was approximately 20 percent with the static opening at position 1 emerging from the medium toward the end of the test.

Tests 6 and 7 (Figs. 9 and 10) were performed on filter C, with the first of these tests using an 80-20 dust mixture, and the second using the same mixture but with 10 percent lint added. In each case the loading was heaviest and the gradient steepest at the points of entrance to the cellular sections of the filter.

Test No. 8 (Fig. 11) was performed on filter D with 100 percent carbon black dust. The resistance gradient curves indicate a reasonably good loading of the dust throughout the filter, but the ineffective packing of the last half inch of the medium is evident throughout the test. The slight negative slope to the gradient lines toward the leaving side of the filter is probably a result of inaccuracies inherent in the method of measurement.

Test No. 9 (Fig. 12) on filter E was performed with an 80-20 dust mixture and 10 percent lint. Again, the pronounced increase in the resistance gradient at the entering surfaces caused by the loading of lint is evident. In this case, no compression whatsoever of the medium occurred and the gradient curves are therefore comparable for each position throughout the test.

In Fig. 13 the data of Fig. 4 are regraphed with the resistance of each quarter-inch layer of filtering medium plotted against the total resistance of the filter. These data would take the same general form if the resistance of each layer of the pack were plotted against time. This method of presentation shows quite clearly how, after the test is partially completed, the dust deposits in one particular layer will become heavier than in other sections of the filter; once this process has started, the resistance gradient at this point becomes progressively steeper. When heavy loading has started at some particular layer the condition becomes progressively worse as the additional load of dust in that section forms

an increasingly efficient area of high resistance into which more and more dust will be deposited. In this case, if the life of the test had been defined by a limiting resistance of 0.4 in. of water, this condition would have been avoided. However, when high limiting resistances are allowed, the condition indicated in Fig. 13 will almost certainly occur at some section, dependent upon the design of the filter and the type of dust to which the filter is subjected. Similar curves may be plotted for the other tests shown in Table 1 and Figs. 4 through 12.

CONCLUSION

The test results indicate that heavy loading of the dust in the filtering medium frequently occurs at some point of one-quarter to one-half inch in depth. The location of this localized high resistance gradient will vary depending upon the nature of the filtering medium and the test dust. Lint normally deposits toward the entering surfaces of the filter and therefore causes localized high resistance gradients at this point. Finely divided carbonaceous dusts deposit more heavily in the more efficient dust filtering areas usually located toward the discharge side of the pack and cause high localized resistance gradients in such sections. Other dusts and mixtures cause steep gradients at intermediate sections.

Any abnormally high localized gradient encountered in the initial unloaded filter may cause excessive initial deposits at this point, and this in turn may eventually result in an extremely steep gradient within a very small depth of the filter. Conditions were encountered in which, at the end of test, as much as 75 percent of the total resistance of the filter could be attributed to localized resistance within a quarter-inch depth of the filtering medium. It is apparent that once a condition of a high localized resistance gradient has developed within a filter, the dust deposits in this section become progressively heavier and in turn result in progressively steeper gradients at this section.

This type of analysis is suggested as a laboratory and possibly a field test. It is evident that, if filters are designed which are not inherently susceptible to the establishing of localized high resistance gradients, the performance characteristics, particularly the life and dust holding capacity of the filter, can be materially improved.

DISCUSSION

R. S. FARR, Los Angeles, Calif. (WRITTEN): Instrument shows points where dust collection results in highest resistance, although not necessarily points of highest dust collecting efficiency and loading. It should be very useful for additional filter design and development as it shows points of major resistance.

The authors state on one type tested that the coarse media and openings in the entering side collect the heavy large particles and the leaving surfaces of denser pack are more efficient in collecting the finer, smaller particles. This indicates that the filter progressively collects finer and finer particle sizes as the air travels from the entering to leaving side. This, of course, could be determined by microscopic examination of dust collected at various points throughout the filter. It is our opinion that if such a gradation of particle sizes trapped in this manner exists, it is due to increased energy picked up by the particles because of the increased velocity caused by the finer pores, for in all cases the particle size collected is many times smaller than the smallest opening in the filter. Our microscopic examination of dust collected on

the entering side of a filter shows particle size distribution proportional to the type of dust fed.

The authors state, in substance, minor alterations of readings may be caused by partial restriction of the static hole opening, necessitating relocation of instrument to get consistent gradient. It is our opinion that partial restriction of opening would not affect reading but merely increase instrumentation lag, as the instrument records only static pressure and not flow. It is our opinion that minor alterations would be the result of velocity head caused by flow non-parallel to the tubes. A possible refinement would be separating the individual tubes and drilling several smaller static holes in each to compensate for variation due to non-parallel flow.

Filters tested are mat type and cellular type. None were tested with primary and secondary air passages. Referring to filters other than the types listed, it may be of interest to know that viscous coated, impingement filters with primary and secondary air passages show a total change in pressure drop of less than 0.1 in. WG when handling even greater dust loads than the filters tested.

The authors are to be commended for the development of a new instrument that should be useful in the design, development, and application of air filters.

A. B. HUBBARD, Bloomfield, N. J. (WRITTEN): The authors are to be commended for their conception of the useful idea of measuring, in detail, the resistance gradients of filters, and for their competence in devising a workable method. Anyone interested in air filters ought to study this carefully and consider the implications with respect to filter design kinds of dust, suitability of test methods, applicability of available filters to a given job, and getting the most out of filters. The authors present their results as a sample of what can be expected of this type of analysis. It is hoped that the sample will be extended in several directions:

1. To include cleanable filters.
2. To develop an ideal reproducible filter which might be used to rate dusts.
3. To make field tests of filters in representative applications.
4. To devise better methods of displaying results so that the implications are more readily apparent.

Cleanable filters could be tested if it were not necessary to sever the filter medium for the insertion of a gradient tube of the type described. It is suggested that a single tube of hypodermic needle size be considered. Such a tube, or needle, could be inserted in almost any filter without preparation. A hole in the side of the needle could be made to register the pressure in successive layers by providing a suitable mechanical system for moving the needle at will. Upstream and downstream pressures could be measured by similar tubes inserted near the moving needle but held stationary.

Comparison of Figs. 4 to 7, inclusive, indicate that each kind of dust can be made to write its own signature. This suggests a possible method of classifying natural dusts by use of an ideal filter devised for reproducibility. It further suggests that the measurement of resistance gradients could show the designer how to manipulate the gradations of the fibers and coatings to fit his product to specific classes of application. A user of filters could likewise profit by resistance gradient studies. The result should indicate choices of filters and methods of maintenance.

In the laboratory testing of filters it is assumed permissible to shorten the test by accelerating the rate at which dust is fed. This assumption would be more acceptible if it could be demonstrated that the gradient curves obtained in a long term field test could be superimposed on those obtained in the accelerated test. Both the rate of feed and the kind of dust undoubtedly would have their effect in such a comparison. It is barely possible that the choice of one could compensate for the influence of the other so that valid laboratory results could be obtained in those cases where the

build-up of gradients matched those obtained in a field test. The comparative study would be worthwhile from any viewpoint.

In connection with the suggestion that improvements are needed in the method of presenting the results, the writer found the following replot of Figs. 4 to 12 helpful: resistance in inches of water was converted to percent of final resistance (at the end of each curve), and the static pressure opening numbers were converted to percent of thickness of the filter. In the cases of Figs. 7 and 8 an attempt was made to allow for compression of the filter medium although this required the, perhaps unwarranted, assumption that all layers were compressed equally. Plotted in this way, the ten curves of a perfect filter would fall on a single line; that is, all elements would be loading equally in terms of the resistance which determines filter life. Any portion of a curve that becomes steeper shows that that layer is either getting more dust or is being compressed. Any portion of a curve that becomes flatter indicates that the corresponding layer is being deprived of opportunity to collect dust, or that it is losing dust (especially if the next layer showed an increase in resistance). These and other effects were made apparent by the percent replots. It was impossible to unscramble portions of the curves in some cases such as Fig. 6. It is believed that if this were done it would show that the 80-20 dust was being collected in the coarse entering layers and then being passed on, in lumps perhaps, to overload layers 4-5 and 5-6. It is further suggested that this effect may be more pronounced in the same filter when it is in actual use and the dust has more opportunity to break loose from one layer to overload succeeding layers.

C. B. Rowe, Madison, Wis. (Written): The authors of Resistance Gradients Through Viscous Coated Air Filters describe a clever approach to obtaining some very valuable filter design information. This information should be useful both for improving the design of a filter media and for evaluating the effectiveness of a particular filter media on a particular type of dust.

It appears that equipment operating on the same principle could be applied to obtaining similar information on permanent type filters.

I would like to ask the authors if they have done any checking to determine whether or not the readings taken at the particular spot where the gradient measuring device is installed are representative of the entire filter.

It is noted that the tests were not terminated until the pressure drop across the filter reached 1 in. of water at 300 fpm air velocity. It is believed that the point of uneconomical operation would be reached long before the total pressure drop reached 1 in. of water. Consequently, I believe the results which are of practical significance are those obtained before the resistance exceeded 0.5 in. of water.

R. S. Dill: There is need for reassurance, I think, that the measuring instrument did not disturb or distort the air filter fibers upon being inserted, with the consequence that the pressures reported may not be truly representative.

R. B. Crepps, Newark, Ohio (Written): The method and apparatus portrayed by Professors Rowley and Jordan point to a successful arrangement for research studies in filter pack design, and their enterprise is to be highly commended.

The authors pointed out limitations in the data presented in this paper; however, the trends, as pictured, may need to be altered if such tool is used for the design studies of any filter pack. It occurs to the writer that the results determined by the use of the special measuring device, placed at one location, may not be representative of the complete action in a particular filter under test. It may easily happen that the tortuous air stream travel through a filter would be altered non-uniformly from point to point as dust particles are collected. Therefore, it would seem appropriate to make concurrent studies at several locations. Furthermore, the authors pointed out that

extreme care needs to be used in the insertion of the special measuring device in order to obtain *true* values, which again indicates that more locations than one should be used in order to arrive at some average which may be representative of the *true* evaluation.

A further interpretation of the pressure gradient would need to be ascertained since dust loading of any particular section might cause a decided emphasis on the shift in position of the original filter layer increments, with respect to the pressure openings in the measuring tool, especially in such cases when the structural integrity of the filter pack would not be ample to keep all pressure openings within the original filter thickness.

AUTHOR'S CLOSURE by PROFESSOR JORDAN: We wish to thank Messrs. Farr, Dill, Hubbard, Rowe, and Crepps for their comments and many valuable suggestions for consideration in any continuation of this test program.

There are several comments which I would like to make in discussion of these suggestions.

First, in answer to Mr. Rowe and Mr. Crepps who have asked the authors if they conducted any tests checking possible variations of resistance gradients at different positions within the filters—we did just that. We ran a series of tests on several filters with the gradient tube located in different positions and found substantially the same results for the initial gradient curves. It is quite possible that at the end of a test there would be more variation in the resistance gradient curves because the dust probably varies not only longitudinally through the filter but also laterally across the face of the filter. I do not feel, however, that this would be of sufficient magnitude to invalidate any of the general results obtained. To a degree variations in total resistance are self-compensating since a point of high resistance will decrease the flow of air and dust until the surrounding sections have built up an equivalent resistance.

Mr. Rowe has suggested that the 1 in. total resistance to which the filter tests were run was of greater magnitude than is usually encountered in the field, and that resistance gradient curves obtained earlier in the tests would probably be more indicative of what would occur in practice.

This is entirely correct. We ran the tests to higher resistances than normally expected in order to accentuate the types of gradient curves found. One inch total resistance is quite high. Although it is sometimes encountered in the field, it is not generally desirable.

The suggestions made as to refinements of the sampling device are entirely in order. This was an initial attempt to propose an additional procedure which might be of value in the running of laboratory filter tests as well as possibly field tests, and we feel that refinements can and should be made if this test is found to be of value.

The *hypodermic needle* suggestion of Mr. Hubbard is an excellent one, and Mr. Farr also made several valuable suggestions concerning apparatus refinements.

Mr. Dill has indicated that he feels that this type of device would destroy the air flow around it. We recognize this possibility and feel that there is some change, as pointed out in the article, of the air flow. However, we do not feel that this is sufficient to invalidate the results in their general magnitude.

Mr. Hubbard has suggested that accelerated tests might be made to show whether the rate of dust feed and the kind of dust would change the characteristics of the filter. We have made and reported much earlier the results of such accelerated tests on efficiency, life, and dust holding capacity of the filter. In these earlier tests we fed dust at rates ranging from 100 grams of dust per hr down to a fraction of a gram per hour and compared the results. The lower concentrations were comparable to those which could be expected in the field in locations where high dust concentrations are encountered.

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No. 1392

FITTING LOSSES FOR EXTENDED-PLENUM FORCED AIR SYSTEMS†

By H. H. Korst*, N. A. Buckley**, S. Konzo^, R. W. Roose††, Urbana, Ill.

Air conditioning systems both summer and winter require the delivery of conditioned air from the unit to the air distributing devices located in the room, such as registers, diffusers, plaques, etc. The branch ducts supplying these air distribution devices commonly consist of round pipes, wall stacks, or rectangular ducts, the sizes of which are determined by the air volume requirements for the space to be conditioned; and the shapes of which are frequently determined by space limitations. The frictional resistances, and pressure losses, of these branch ducts and fittings have been fairly well established.[1, 2, 3]

The trunk which connects the branch ducts to the conditioner, or to the plenum attached to the conditioner, can consist of:

1. A non-uniform size arrangement in which the cross-sectional area is decreased following each branch take-off connection. As shown in Fig. 1*b* this arrangement requires several different trunk sizes and a number of complicated fittings.
2. A smaller, uniformly sized plenum, shown in Fig. 1*c* which is designated as an extended plenum. In this case the cross-sectional area of the plenum is approximately equal to that of the inlet. This arrangement consists of a single size of trunk and relatively simple branch take-off fittings. Since a single size trunk is used on a given installation and a small number of fittings is required, the installation is relatively simple and less costly than the more complicated arrangement shown in Fig. 1*b*.
3. A large, uniformly sized plenum, such as is shown in Fig. 1*d*, and which has been designated as a box plenum. This type of plenum refers to that having a large

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† This investigation is a part of the cooperative project jointly sponsored by the Engineering Experiment Station of the University of Illinois and the *National Warm Air Heating and Air Conditioning Association*.

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[1] Exponent numerals refer to References.

Presented at the 56th Annual Meeting of The American Society of Heating and Ventilating Engineers, Dallas, Tex., January 1950.

cross-sectional area relative to the inlet area. The box is of rectangular construction without transition sections, and the branch take-off fittings consist of simple, butted connections. Pressure loss and air flow characteristics of a box plenum have been reported.[4]

However, information dealing with the pressure loss and air flow characteristics of the extended plenum type of duct system has not been available. The present investigation, therefore, has been devoted to a study of the extended plenum and its take-off fittings. The emphasis has been largely devoted to the size and shape of the junction or take-off fitting between the extended plenum and the branch duct, since the fitting resistance can be a substantial portion of the total resistance of the system when unfavorable forms are used. The objective of this investigation, therefore, was to determine shapes of take-off fittings which not only would give good performance, but also would be simple in construction. In view of the fact that an unlimited number of combinations of branch ducts and extended plenum are possible, it was necessary in this preliminary study to confine the investigation to a single arrangement of branch and plenum. For this purpose a single branch duct was connected to a 12 in. × 8 in. extended plenum. A large number of branch take-off fittings were tested over a range of air quantities, and pressure losses were determined across the branch fitting. Losses in the trunk duct across the take-off fitting were also studied but have not been included in this paper.

Analysis of Tests Made by Previous Investigators

The losses at a junction between a trunk duct and a branch duct can be considered as being composed of both friction losses and mixing losses, not only for the flow in the branch duct but also for the flow in the trunk duct. Of the two component losses, the mixing losses usually dominate, and occur when an unstable flow pattern changes to a stable velocity distribution as a result of internal friction. For example, if the flow in the branch duct is abruptly or sharply turned in direction, the inertia of the flowing mass will produce a separation of the stream from the wall with the resultant formation of a vena-contracta section at some distance downstream. The stabilization of this non-uniform velocity profile is accompanied by a mixing loss. In the same manner, the diversion of part of the flow from the trunk duct into a branch duct, results in a disturbed velocity profile in the trunk duct, and the subsequent stabilization of flow in the trunk duct is accompanied by a mixing loss.

If the flow pattern for a given fitting were exactly known, or could be predicted, a computation of the loss could be made by means of the momentum equation. For example, in the case of a water jet issuing into air in which a sharply defined boundary exists between the two fluids, the free-streamline method introduced by Helmholtz and later used for special problems by Kirchhoff, von Mises,[5] and others could be used in the analysis of the flow at the junction of the jet and the pipe. However, the method is limited in application to rather simple two-dimensional cases involving sharply defined boundaries of the issuing jet, and for constant pressure existing at the free surface. The analytical approach proves inadequate in those cases where the jet surface is not sharply defined, and an unstable discontinuity in the fluid disintegrates into eddies, such as is the case when an air jet issues into air, or when a water jet issues into water.

An analytical method was used by Vazsonyi,[6] in which the equation of motion was replaced by the momentum equation for the solution of problems dealing with flow in miter elbows and branch take-off fittings from a trunk duct. The assumption was made not only that the velocity profile was fairly equalized across the duct, but also that the pressure distribution across the containing walls of the duct was equal. Since considerable uncertainty exists in these basic assumptions, any analysis of this nature must be modified by correction factors determined from actual tests.

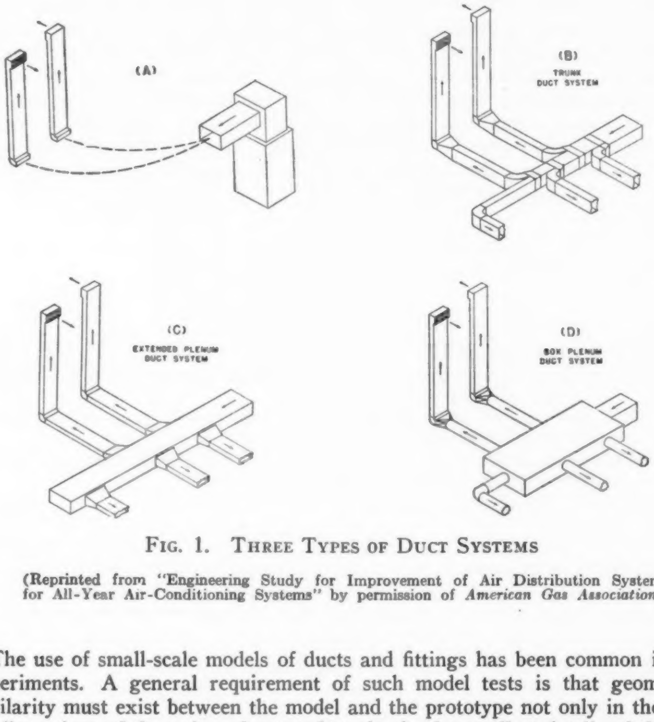

FIG. 1. THREE TYPES OF DUCT SYSTEMS

(Reprinted from "Engineering Study for Improvement of Air Distribution System for All-Year Air-Conditioning Systems" by permission of *American Gas Association*)

The use of small-scale models of ducts and fittings has been common in flow experiments. A general requirement of such model tests is that geometrical similarity must exist between the model and the prototype not only in the overall dimensions of the unit under test, but also in the smallest details of the unit. For example, the relative roughness of the walls must be the same for both model and prototype. The results of tests on the models made under these conditions can then be extrapolated to conditions anticipated for the prototype by the application of a correlation number or model law. For the case under consideration in this paper in which the influences of compressibility and heat transfer are negligible the Reynolds correlation would be applicable. However, as will be discussed later, the results of previous investigations[7, 8, 9, 10, 11] seem to indicate that even the Reynolds number has but little significance as far as the junction losses are concerned.

Previous tests on junction losses, such as those reported by Thoma, director, Munich Hydraulic Institute, and his associates[7, 8, 9, 10] have been confined to round branch pipes and were made with water. The test installation, which has been described by Petermann,[9] consisted of a trunk 43 mm in diameter (approximately 1.7 in.) and three separate branch pipes, 15 mm, 25 mm, and 43 mm. The branch junction was located approximately 47 diameters from the intake section of the trunk. The upstream measuring station was about 38 diameters from the intake section and sufficiently far from the intake to obtain a stable velocity profile. The downstream measuring station in the trunk was 63 diameters from the intake. The downstream measuring stations in the branch were from 45 to 75 diameters downstream from the junction, and were also located at sections having a stabilized velocity profile. The difference in total head at the upstream and downstream measuring stations for both trunk and the branch pipe included, therefore, some amount of duct friction head. Since it was not possible to evaluate this duct friction head under actual conditions of test, a separate test of a straight pipe section was made. The *net* junction losses were obtained by subtracting this nominal pipe friction from the total head, and were then expressed in terms of the velocity head.

That is,

$$\lambda_u = \frac{(TP_1 - TP_2) - SP_f}{VP_u}$$

in which,

λ_u = the coefficient of loss.
TP_1 = total pressure measured in the trunk duct upstream of the take-off fitting, in. of water.
TP_2 = total pressure measured in the branch duct downstream of the take-off fitting, in. of water.
SP_f = the static loss due to friction in the trunk and branch ducts between the measuring stations and the fitting, in. of water.
VP_u = the velocity pressure in the trunk duct, in. of water.

The evaluation of the nominal pipe friction of a straight pipe has no actual significance, but is of importance in practical applications of the results, since the computation of the friction losses for the entire duct system can then be made by adding the losses for the component parts of the entire duct system.

The results obtained by Thoma indicated that the junction losses, λ_u, thus obtained were functions only of the ratio, Q_b/Q_u, in which Q_b and Q_u represented the flow volumes in the branch pipe and united flow in the trunk, respectively. Apparently, the junction losses were not dependent upon the mass rate of flow; in other words, the losses were independent of Reynolds number, N_{Re}. The test results reported in references 7 and 10 have been replotted as shown in Fig. 2 to show the junction loss, λ_b, expressed in terms of the velocity head in the branch pipe. The curves show not only the comparative effects of angle branching and right-angled tees, but also that the junction loss, λ_b, is proportionately greater for smaller diversions of the flow into the branch pipe than for large diversions. As anticipated, angle branching appears to be an effective method for reducing junction losses. In addition, the use of a conical intake is effective in reducing the losses of a right-angled junction, especially for higher ratios of Q_b/Q_u.

The curves in Fig. 2 were all for a ratio of branch diameter to trunk diameter, d_b/d_u, of 25/43. If the test results for different diameter ratios had been plotted on the same figure, a separate and distinct curve would be obtained for each diameter ratio. However, as suggested by Brabbee[12] and Vazsonyi,[6] by plotting the coefficient of loss against the independent variable of V_b/V_u instead of Q_b/Q_u, it should be possible to present the results obtained with different diameter ratios by a single curve. This has been done by replotting the loss

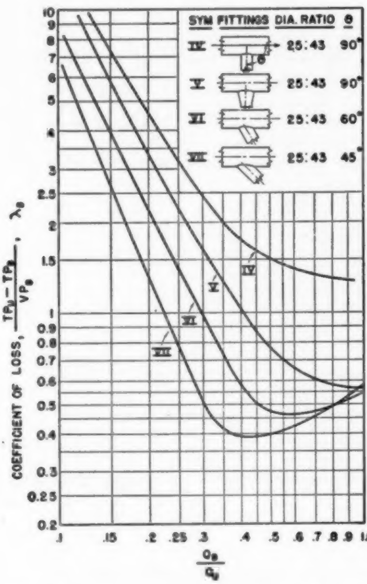

FIG. 2. COMPARISON OF FOUR TYPES OF FITTINGS TESTED BY THOMA AND ASSOCIATES

(Data replotted from References 7, 9, 10)

coefficients obtained by Kinne[10] for a 60-deg angle branch having diameter ratios of 15/43, 25/43, and 43/43. As shown in Fig. 3, the agreement is fair since the separate curves do not deviate too far from each other. Additional tests of other types of junctions would be desirable to establish the fact that a single curve would represent results for a wide range of diameter ratios.

Another investigation of junction losses was made in connection with model tests of penstocks for the Boulder Dam project.[11] Since the magnitude of N_{Re} for the Thoma tests was of the order of 100,000 while that for the prototype junctions of the Boulder Dam project was of the order of 90,000,000, tests were made with models for which the range of N_{Re} was between 130,000 and 800,000.

Some special tests with a 90-deg junction were made in an attempt to duplicate the work of Vogel[7] under conditions of a larger scale and a higher N_{Re}, but the coefficients of loss were found to be considerably lower in magnitude. Another series of tests made with a slightly different junction showed, however, no influence of N_{Re}. If this is also true for the 90-deg junction, the discrepancy in the coefficients of loss between the Boulder Dam and the Vogel tests is difficult to explain. Hence, the relationship between the coefficient of loss and N_{Re} cannot be considered as definitely established.

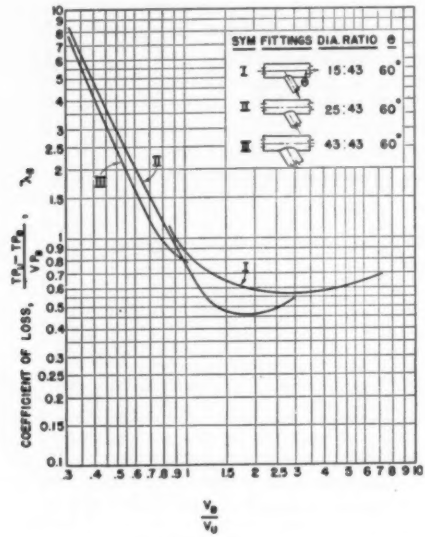

FIG. 3. AREA RATIO EFFECT ON COEFFICIENT OF LOSS WHEN PLOTTED AGAINST VELOCITY RATIO

(Data replotted from Reference 9)

In the report of the Boulder Dam tests, an empirical equation of the following form was given as representing the measured losses of the junction:

$$\lambda_b = K + K_1 \ (V_u/V_b)^2$$

in which, K is a normal entrance-loss coefficient, and K_1 is an empirical constant.

The present writers were interested, however, in determining whether the application of the Borda-Carnot equation to the published data on pressure readings would yield results of any value. For this analysis the assumptions were made that: (*a*) the velocity across the vena contracta section was uniform, (*b*) the losses between the trunk duct and the vena contracta section were negligible, and (*c*) that the mixing loss downstream from the vena contracta section could

be obtained from the Borda-Carnot equation. The results of this analysis were in substantial agreement with the experimental data, especially for higher ratios of Q_b/Q_u. This indicates in general that the mixing losses are a dominant part of the entire junction loss.

In general, the study of work by previous investigators indicated the necessity of actual tests of any new junction form, preferably with full scale equipment.

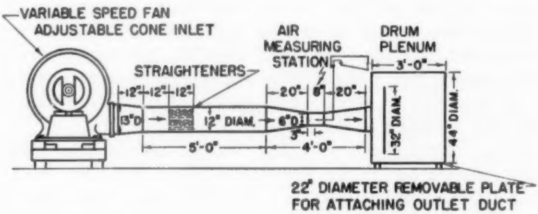

FIG. 4a. TEST ARRANGEMENT FOR MEASURING AND DELIVERING AIR

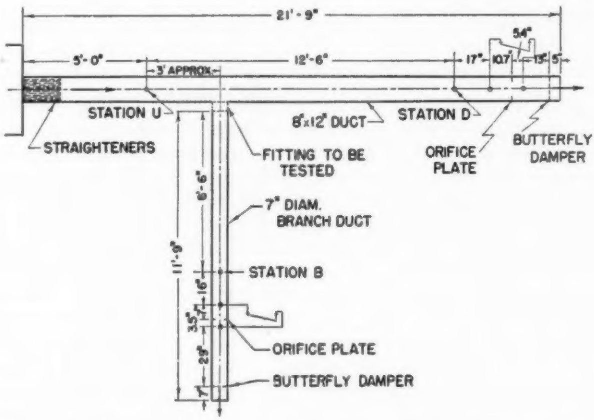

FIG. 4b. TEST ARRANGEMENT SHOWING DRUM PLENUM, EXTENDED PLENUM TRUNK DUCT, AND FITTING

DESCRIPTION OF EQUIPMENT

The arrangement for metering the air and for delivering the metered quantity of air to the test section is shown in Fig. 4*a* and has been described in a previous publication.[4]

In Fig. 4*b* the 12 in. × 8 in. extended plenum trunk duct is shown attached to the drum plenum, and the fitting to be tested was attached to either the top or side

of the trunk duct approximately 8 ft. from the face of the drum plenum. Egg-crate straighteners installed in the 12 in. × 8 in. duct, at the point where the trunk was attached to the plenum, were for the purpose of eliminating the swirling motion of the air stream. Piezometer connectors were installed on the trunk duct 5 ft and 17½ ft from the face of the drum plenum and in the branch duct 6½ ft from the trunk duct. These piezometer connectors were fastened at the center of the top, bottom, and both sides of the ducts in the same vertical plane, and were then connected in parallel to give an average reading of pressures.

For measuring the air flow in the trunk duct downstream of the branch, an orifice with an open area of 50 percent, together with diameter taps, was installed in the trunk duct approximately 20 ft from the face of the drum plenum. Flow through the orifice was controlled by a butterfly damper installed at a sufficient distance downstream to avoid interference. A similar orifice and damper arrangement was installed in the branch duct.

The fittings tested were of three distinct types. The side take-off fitting placed over an opening in the side of the trunk duct, delivered air to a branch duct in the same horizontal plane as the trunk duct. A second type of side take-off fitting employed an expansion section inserted into the trunk duct with a second fitting to turn the air into the branch duct in the same horizontal plane. Top take-off fittings were placed over an opening in the top of the trunk duct and delivered air to a branch duct running perpendicular to the trunk duct but in a horizontal plane above the trunk duct.

CALIBRATION OF EQUIPMENT

For purposes of calibration the arrangement shown in Fig. 4*a* was used. Three ducts consisting of a 12 in. × 8 in. rectangular trunk duct, a 7 in. diameter round pipe, and a 5 in. × 8 in. duct were separately connected to the drum plenum shown in Fig. 4*a* for purposes of calibration. For all three duct arrangements, egg-crate straighteners and piezometer rings were installed. The upstream piezometer ring was located about five feet from the plenum, and the downstream piezometer ring was about 17½ ft from the plenum. A series of tests was conducted to compare the pressure readings of the piezometer rings with those obtained by means of a traversing static pressure tube. The pressure holes in this traversing tube were located in the plane of the piezometer ring. With a definite air quantity flowing in the duct, a twenty-point equal-area traverse was made with this tube located at both stations, and simultaneous readings were made with the piezometer ring. The separate pressure measurements obtained with these two devices were found to be in excellent agreement. Hence, in subsequent tests the readings obtained by means of the piezometer ring were accepted as a true index of the static pressures at the two measuring stations.

For the purpose of determining the friction loss in the three separate test ducts, a series of tests was conducted, during which static pressures were observed at the two piezometer rings. The friction loss per 100 ft of straight duct could be evaluated for a given air quantity. The data obtained from these tests were compared with similar data reported by D. K. Wright, Jr.[13] In general, the Illinois data were in fair agreement with the Wright data, but since the former data were derived specifically for the ducts used in this investigation, they were selected for later use.

A sharp-edged orifice with radius taps was installed approximately 2 ft downstream from the second piezometer ring. The area ratio of duct to orifice was approximately 2 to 1. With this arrangement, calibration tests were conducted, during which the air quantities flowing at the air measuring station and the pressure differential across the orifice were simultaneously determined. From these tests, final calibration curves were established for the orifices installed in the three separate test ducts. These orifices provided a means in later tests of evaluating the separate air quantities flowing in the extended plenum and branch ducts.

Procedure

The procedure for testing all types of fittings was the same. With the fitting installed in the test arrangement shown in Fig. 4*b* the total air quantity flowing at the air measuring station upstream of the drum plenum was set at the desired quantity and held constant throughout the test. Then air quantities flowing in the trunk and branch duct, and static pressure readings at stations U, D, and B were observed. The quantity of air flowing through the branch duct was varied from zero to 100 percent of total flow by varying the setting of the butterfly dampers in the trunk and branch ducts. For each damper setting, a complete set of readings was recorded. For example, in the test for the butt take-off fitting with 555 cfm flowing in the system, and with both the damper in the trunk and the damper in the branch duct wide open, the air quantity flowing through the trunk duct was 415 cfm and through the branch duct 140 cfm. The static pressure readings at stations U, D, and B were 0.091, 0.100, and 0.062 in. of water, respectively.

Test Results

90-Deg Side Take-Off Fittings. The test results for 90-deg side take-off fittings are shown in Fig. 5. Following the usual procedure the coefficient of loss, λ_b, is presented as a function of the quantity ratio, Q_b/Q_u. The coefficient of loss is herewith defined as:

$$\lambda_b = TP_u - TP_b/VP_b$$

in which,

λ_b = the coefficient of loss in branch velocity heads.
TP_u = the total pressure at station U, in. of water.
TP_b = the total pressure at station B, in. of water.
VP_b = the velocity pressure at station B, in. of water.
Q_u = the air quantity flowing at station U, cfm.
Q_b = the air quantity flowing at station B, cfm.

Fitting 3, which is a butt take-off, is comparable to that used[7] in determining curve IV of Fig. 2. It may be noted from Fig. 5 that fitting 3 gave relatively high coefficients of loss. In attempting to improve on this butt take-off connection, a number of modifications were tested of which fittings 23, 24, and 25 demonstrate the major elements that affect the coefficient of loss. These three fittings had a 12 in. × 7 in. inlet opening and a 7 in. round outlet and were installed with the 12 in. dimension along the trunk duct. The center lines for the branch duct and trunk duct were in the same horizontal plane. However,

the center of the 7 in. round outlet was shifted relative to the center of the 12 in. dimension of the inlet.

Fitting 24 approached a symmetrical design with the 7 in. round opening shifted 1 in. in the downstream direction. It is apparent that the conical entry decreased the coefficient of loss over the complete range of quantity ratios.

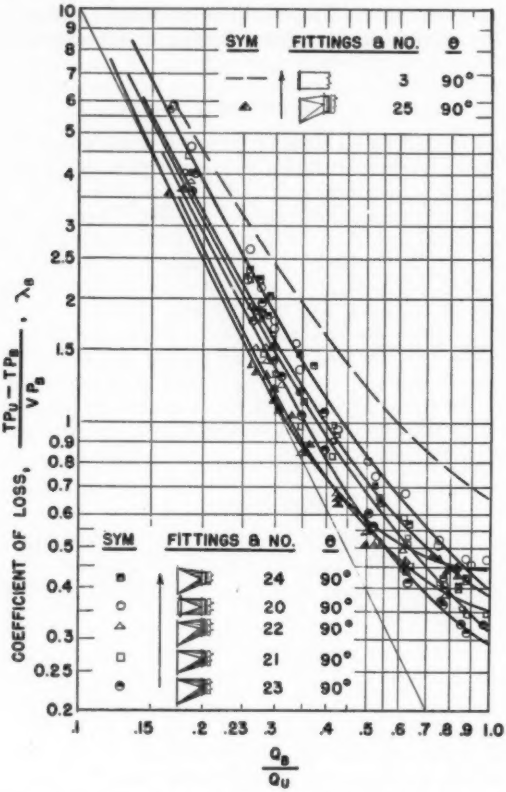

FIG. 5. LOSS CURVES FOR 90-DEG SIDE TAKE-OFF FITTINGS

In fitting 23, the 7 in. round opening was shifted 2½ in. in the downstream direction, thus reducing the sharpness of the turning angle at the upstream entrance of the fitting. A reduction in coefficient of loss for comparable quantity ratios was again obtained. This was attributed to the reduction in separation of flow from the walls following the upstream edge of the fitting.

A further shifting of the 7 in. round opening was tested in fitting 25 for which the round opening was shifted 4 in. in the downstream direction. For this

arrangement a reduction in coefficient of loss was obtained for lower quantity ratios. However, for higher quantity ratios the coefficient of loss tended to approach a constant value, which was attributed to a second separation at the downstream edge of the inlet opening. For example, when all of the air was flowing out of the branch duct, ($Q_b/Q_u = 1.0$), it is probable that the direction of flow at the entrance of the fitting was nearly at right angles to the trunk duct wall, resulting in a second separation at the downstream edge of the inlet opening. Fittings 22, 23 and 25 gave the best results, indicating the desirability not only of using a conical entry, but also of reducing the sharpness of the

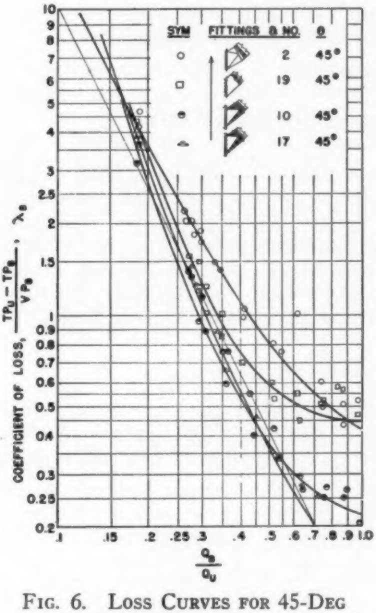

FIG. 6. LOSS CURVES FOR 45-DEG SIDE TAKE-OFF FITTINGS

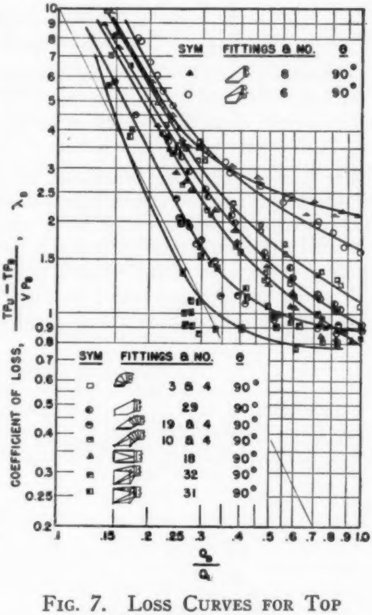

FIG. 7. LOSS CURVES FOR TOP TAKE-OFF FITTINGS

turning angle at the entrance to the fitting. Although the coefficient of loss for fitting 25 at high ratios of Q_b/Q_u was larger than those for either fittings 23 or 24, the magnitude of the differences was negligible.

Angle Take-Off Fittings. Results obtained with angle take-off fittings are shown in Fig. 6. Fitting 19 is a 45-deg angle branch without conical entry, and the results are closely comparable to those shown by curve VII in Fig. 2. Improvement was again obtained for conical entry as shown by fittings 10 and 17.

Top Take-Off Fittings. Results obtained with top take-off fittings are shown in Fig. 7. Top take-off fittings necessitate two turns in two different planes, which tend to produce rotational flow. Rotational losses are extremely high

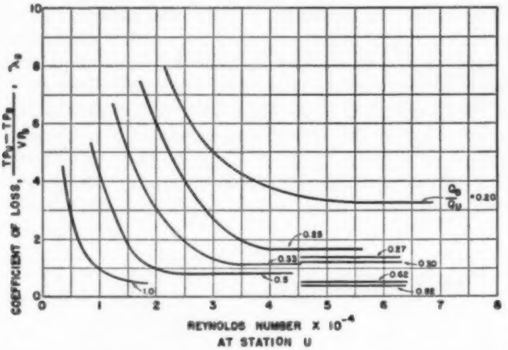

FIG. 8. LOSSES AS RELATED TO REYNOLDS NUMBER

when a contraction or separation of flow occurs. Fitting combination 3 and 4 which consisted of a butt take-off on the top of the trunk duct immediately followed by a 90-deg elbow gave poor results. Although rotation cannot be avoided when a top take-off fitting is used, separation can be reduced by measures similar to the conical entries used previously for side take-offs. Fittings 31 and 32, which incorporated both conical entry and reduction in sharpness of turning angle at the entry of the fitting, gave the best performance of those tested.

REYNOLDS NUMBER INFLUENCE

Coefficient of loss λ_b has been presented in Figs. 5, 6, and 7 as depending only upon the quantity ratio Q_b/Q_u. This is true only if the Reynolds number, N_{Re}, has no influence. As explained earlier, the findings of previous investi-

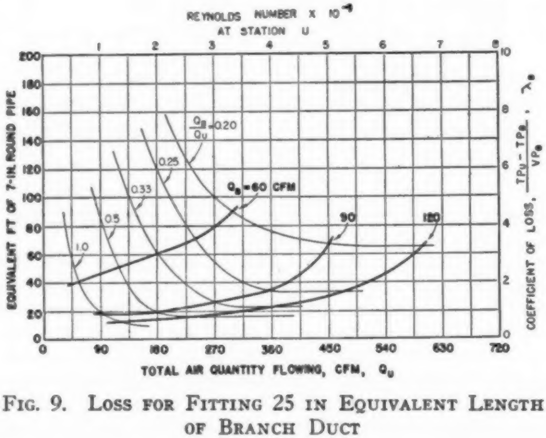

FIG. 9. LOSS FOR FITTING 25 IN EQUIVALENT LENGTH OF BRANCH DUCT

gators[7, 11] were not conclusive, and since the curves for coefficient of loss for this project were determined for total air volumes of 425 and 550 cfm corresponding to the Reynolds numbers of 5.56×10^4 and 6.11×10^4, respectively, it was necessary to explore a wider range of Reynolds Numbers.

In this discussion, Reynolds Number is defined as:

$$N_{Re} = 4mV\rho/\mu$$

in which

N_{Re} = the Reynolds number for the trunk duct.
m = hydraulic radius of the trunk duct, feet.
V = the velocity in the trunk duct, feet per second.
ρ = the density of the fluid, pounds per cubic foot.
μ = absolute viscosity, pounds per foot-second.

A special series of tests was conducted with fitting 25 for total air quantities ranging from 40 cfm to 600 cfm, corresponding to N_{Re} from 4.43×10^3 to 6.66×10^4, respectively. The results of these tests are plotted in Fig. 8. For a particular quantity ratio, the coefficient of loss is constant for large values of N_{Re}, but increases for small values of N_{Re}. Therefore, a limitation exists in the method of plotting used in Figs. 5, 6, and 7, in which the coefficient of loss is considered to be a function of the quantity ratio only. For very low values of N_{Re}, that is, for low total air quantities flowing, an increase in coefficient of loss can be expected, which in turn will reduce the applicability of the test data illustrated in Figs. 5, 6, and 7.

Therefore, the results shown in Figs. 5, 6, and 7 hold only for conditions in which N_{Re} has little or no influence, which limits the application of these test results.

PRACTICAL INTERPRETATION OF TEST RESULTS

For practical application, the methods of presenting results given in Figs. 5, 6, 7, and 8 were not convenient to use. A method of presenting these data in terms of actual air volumes rather than N_{Re}, and equivalent lengths rather than λ_b would be of more practical value. For this purpose the values for Fig. 8 were replotted as shown in Fig. 9. For the given system in which the trunk duct was 12 in. × 8 in. and the branch duct was a 7-in. round pipe, and the air properties were considered as remaining constant, it is possible to replace the N_{Re} in Fig. 8 by air quantities in cfm. Furthermore, with total air quantities established, it is possible to determine curves of constant branch quantity for the particular branch size. Finally, it was also possible to convert the coefficient of loss for a given fitting into terms of equivalent length of branch duct. In this conversion, the friction factor, f, was considered to be constant, which is equivalent to applying a constant conversion factor between the coefficient of loss and the equivalent length of branch duct.

The method of representing data which is shown in Fig. 9 is the most convenient for practical application, but would require a separate set of curves for each combination of branch and trunk duct to be used. However, this figure demonstrates that:

1. A constant value of equivalent length cannot be expected.
2. The take-off nearest the conditioner will have the largest equivalent length, as compared to a second, third, or other branch, when equal conditions of branch size and air delivery are maintained.

Equivalent length values were established for four representative fittings tested and were plotted in Fig. 10. As mentioned previously, these equivalent lengths apply directly only for the arrangement given and in the region where the fittings loss is independent of N_{Re}.

COMPARISON WITH PREVIOUS RESULTS

The fittings used in this investigation were not geometrically similar to those used by previous investigators, so that a direct comparison of results was not

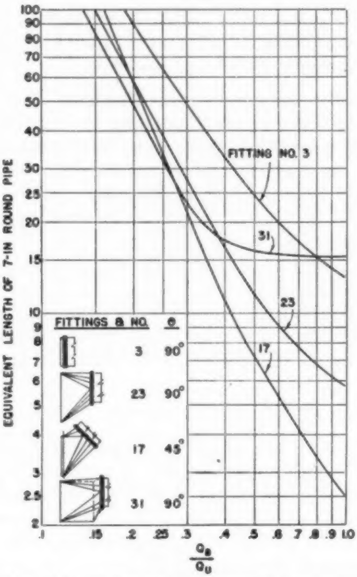

FIG. 10. LOSS OF FITTINGS EXPRESSED IN TERMS OF EQUIVALENT LENGTHS OF BRANCH DUCT

possible. However, fitting 19 corresponded closely to the 45-deg branching angle tested by Petermann.[9] In the latter tests, water flowed through a 43 mm trunk and a 25 mm branch pipe. Petermann also conducted tests with ratios of branch diameter to trunk diameter of 15/43 and 43/43. These data are shown in Fig. 11 and are represented by solid lines. In addition, a curve for coefficient of loss is shown for fitting 19, for which the branch diameter was 7 in. and the equivalent diameter of the 12 in. × 8 in. trunk based upon equal areas, was 11.05 in. In terms of the diameter ratios used by the previous investigators, this fitting corresponded to a ratio of 27.25/43, and the results obtained are shown in Fig. 11 by a broken line. It may be noted that the location of the curve for fitting 19 was fairly consistent with respect to the others, considered on the basis of the variation in diameter ratios.

A further analysis was made of the results of these three separate projects, by plotting the coefficient of loss against V_b/V_u, as was previously discussed in connection with Fig. 3. The losses for the four separate fittings having different area ratios could be represented by a single curve. This indicated that for a given type of fitting, a wide variation in the ratio of cross-sectional areas of branch and trunk duct would yield results which can be represented by a single curve based on a V_b/V_u scale. Furthermore, if a similar relation holds

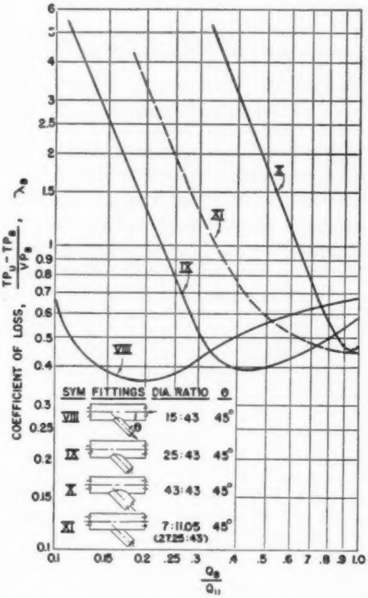

FIG. 11. COMPARISON OF DATA FROM ILLINOIS AND MUNICH

(Munich data replotted from Reference 9)

true for other types of fittings, then a family of curves similar to those shown in Fig. 11 can be established for each fitting. Tests could then be made of a particular combination of branch fitting and trunk duct, and the results extrapolated to any other combination of sizes of branch and trunk ducts. Additional tests to confirm this conclusion would be desirable.

CONCLUSIONS

Under the conditions maintained in these tests the following conclusions were made:

1. The results of this investigation were in substantial agreement with those of previous investigators.

2. Butt take-off fittings connecting a branch pipe to an extended plenum gave relatively high coefficients of loss for both side and top take-offs.

3. Conical entry decreases loss through both angle branches and 90-deg take-off fittings.

4. Reduction in sharpness of the turning angle at the entry of a fitting decreases loss through the fitting.

5. For large values of Reynolds number, *i.e.*, for large air quantities flowing the coefficient of loss is practically constant; however, for very low values of Reynolds number an increase in coefficient of loss can be expected.

6. A constant value of equivalent length cannot be designated for a particular fitting, but depends upon the quantity ratio, and to some extent upon the total air quantities.

ACKNOWLEDGMENT

This paper is a report of an investigation conducted under the terms of a cooperative agreement between the *National Warm Air Heating and Air Conditioning Association* and the Engineering Experiment Station at the University of Illinois in the Mechanical Engineering Laboratory. The results presented in this paper will be ultimately included in a bulletin of the Engineering Experiment Station. Acknowledgment is made to the Research Advisory Committee of the Association for advice and counsel received.

This paper includes material from two theses completed under the direction of Prof. S. Konzo and submitted in partial fulfillment of the requirements for the degree of Master of Science in Mechanical Engineering, by E. R. Zieve in 1947 and J. J. Boland also in 1947.

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DISCUSSION

D. W. LOCKLIN, Cleveland, Ohio (WRITTEN): The Technical Advisory Committee on Air Distribution is interested in the overall problem of losses in duct fittings as part of its long-range program of research. A literature survey* of available information on losses in elbows as one type of *through-flow fitting* is under way at the Laboratory; this paper is a contribution in the field of losses in *branch fittings.* Study of the paper has led to the following comments:

1. *Fitting Dimensions:* More completely dimensioned drawings of the fittings tested would add to the value of the paper. For example, the lengths of conical section for the fittings of Fig. 5 are not indicated.

2. *Trunk Loss:* The authors have implied that the losses in the trunk duct (between stations U and D Fig. 4) will be reported later. These data are of interest, especially since few data are available on this phase of duct system analysis.

3. *Reynolds Number:* The limitation of the plotting methods exemplified by Figs. 2, 3, 5, 6, 7, 10, and 11 have been emphasized by the authors due to the Reynolds number effect indicated by Fig. 8. The Reynolds number characteristic, however, is probably different for each fitting; it is suggested by the data of Weske† that the characteristic may even show a rising loss at higher Reynolds number in some fittings.

4. *Friction Factor:* The authors have employed a constant friction factor of 0.0292 in converting the coefficient of loss for fitting 25 in Fig. 8 to equivalent feet of branch pipe as shown in Fig. 9. An attempt has been made by the writer to determine the effect of making this conversion using a friction factor which varies with Reynolds number in the branch. These conversions were made on the basis of the friction factors, f, proposed by Moody‡ and the absolute duct roughness of 0.0005 ft adopted by Wright for the A.S.H.V.E. Friction Chart (see Reference 13, p. ?).

The data so converted are plotted in Fig. A showing a comparison with the data given in the paper. At the lower range of Reynolds number for each of the diversion ratio curves shown in Figs. 8 and 9, the equivalent length using $f = 0.0292$ is approximately 20 percent higher than when the variable values are used, and approximately 10 percent lower at the higher Reynolds numbers; the curves cross when the friction factors are coincident, this corresponding to a branch flow of about 77 cfm. For low total quantities of air flowing or for small diversion ratios Q_b/Q_u the Reynolds number in the branch approaches the critical zone‡ and yields a friction factor which increases quite rapidly with decreasing Reynolds number, accounting for a relatively large range of branch friction factors for this investigation.

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* A.S.H.V.E. RESEARCH REPORT No. 1405—Energy Losses in 90-Degree Duct Elbows—A Survey and Analysis of Available Information, by D. W. Locklin (A.S.H.V.E. TRANSACTIONS, Vol. 56, 1950, p. 479).

† Pressure Loss in Ducts with Compound Elbows, by J. R. Weske (*National Advisory Committee for Aeronautics, Advanced Restricted Report* W-39, February 1943).

‡ Friction Factors for Pipe Flow, by L. F. Moody (*A.S.M.E. Transactions,* Vol. 66, 1944, p. 671).

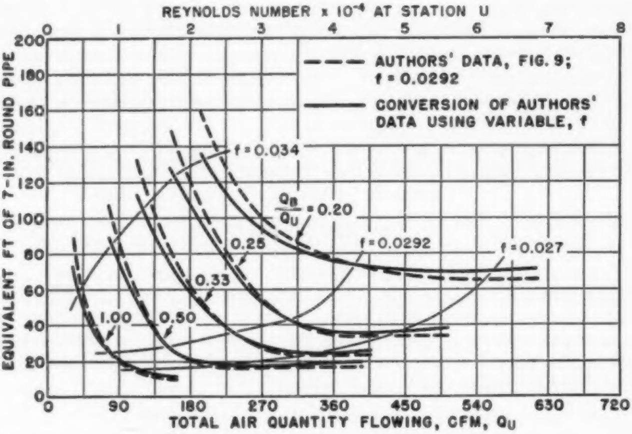

FIG. A. COMPARISON OF EQUIVALENT LENGTH CONVERSION METHODS

For residential design work such a refinement may not be justified; however, the difference is significant and warrants consideration.

AUTHORS' CLOSURE: An attempt has been made by the authors to describe the typical trends of the flow in branch fittings, to show the ways for possible improvement in performance and to give a suitable method of analysis as well as rational forms for the presentation of data. More complete information, as needed for practical application in design will be furnished in a final report of the Engineering Experiment Station of the University of Illinois, which will include data on the trunk duct losses.

The discussion of the influence of Reynolds number is necessary as it shows the limitations for a presentation of the dimensionless loss coefficient as a function of the ratio of the flow rates only. This was done, as an illustration, for one fitting only, namely No. 25. It appears clearly in Fig. 8 that a difference exists between a domain of no influence Reynolds number and another where this does not hold true. That the limit between the two domains falls into a range of extreme low flow rates as far as practical applications are concerned, is shown in Fig. 9.

It can be expected that this lower limit for the validity of an exclusive λ *vs* Q_b/Q_u presentation, although obtained for one fitting only, allows a more general interpretation: No. 25 has a favorable performance characteristic due to the fact that a conical intake was used in order to achieve a smooth turn. For this fitting an increase for the loss coefficient is observed when the local Reynolds number in the conical intake is of the order of the critical value. Since friction forces are the reason for an influence of Reynolds number and become a factor when the dominating influence of inertia (separation) losses decreases, fittings with less favorable flow conditions than No. 25 should give loss coefficients dependent upon the flow ratio only, to even lower values of Reynolds number as it is defined in the paper for the trunk flow.

No. 1393

VANEAXIAL FAN FUNDAMENTALS†

By Raymond Mancha*, Pittsburgh, Pa.

General Description

The principal parts of the conventional vaneaxial fan equipped with stationary vanes consist of the *inlet section, stage* and *stack* as illustrated by Fig. 1.

Inlet section is equipped with a cowling and cowling supports and is designed so as to permit air to enter the stage with the airflow pattern upon which the design depends.

Fan *stage* consists of the rotor and stationary vanes; it is within the stage that the air is compressed and delivered to the stack with the proper airflow pattern to permit efficient stack action.

Stack is equipped with a discharge cowling and cowling supports and is designed to efficiently convert the air speed from the fan outlet to the desired value. Thus, a blowing fan can supply air to the system at the desired air speed and an exhaust fan can discharge air to the atmosphere with low kinetic or waste energy.

Capacities and Speeds

For a particular size vaneaxial fan equipped with stationary vanes, customary design practice dictates a design air volume and a design fan speed for any particular fan pressure. When passing the design air volume, conventional design provides that the air enters and leaves the fan stage with equal stage pressure increase and with pure axial air flow of uniform and equal speed at all radii. Design speed is the fan speed corresponding to design air volume at design fan pressure.

For geometrically similar fans, design air volume varies directly as the square root of design fan pressure and directly as the square of the fan diameter. Like-

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† *Abstract:* The development of formulas and the details of design are described more fully in the Appendix.

* Vice president, Ventilation, Joy Mfg. Co.

Presented at the 56th Annual Meeting of The American Society of Heating and Ventilating Engineers, Dallas, Tex., January 1950.

wise, design fan speed varies directly as the square root of design fan pressure but inversely as fan diameter. For classifying a family of geometrically similar fans as to capacity and speed, it is therefore convenient to refer to a one foot diameter prototype handling air weighing 0.075 lb per cu ft and developing a fan pressure of one inch water column. Thus, design air volume is called design unit volume, and design fan speed is design unit speed.

General classification of vaneaxial fans as to capacity or speed is entirely arbitrary. However, it might be useful to refer to design unit volumes of 1000 cfm or under as low capacity, from 1000 cfm to and including 2000 cfm as inter-

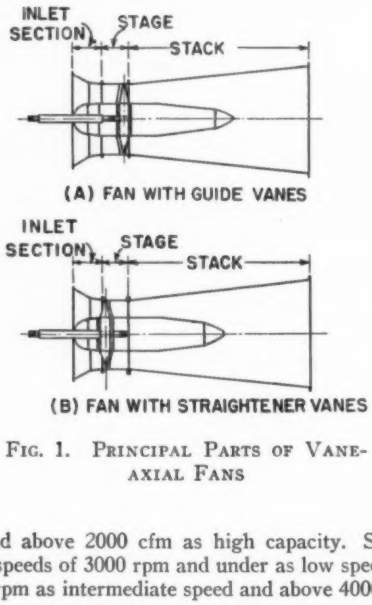

FIG. 1. PRINCIPAL PARTS OF VANE-AXIAL FANS

mediate capacity and above 2000 cfm as high capacity. Similarly, one might classify design unit speeds of 3000 rpm and under as low speed, from 3000 rpm to and including 4000 rpm as intermediate speed and above 4000 rpm as high speed.

The ratio of the fan rotor diameter at the blade root to the rotor diameter at the blade tip is referred to as hub ratio. This ratio is preserved throughout the size range of any family of geometrically similar fans.

ROTOR ACTION

The total work done upon the air passing through a vaneaxial fan rotor is partially realized by a mechanical energy addition to the air equal to the algebraic sum of the static pressure energy increase and the rotative energy decrease or increase, depending upon the location of the stationary vanes.

With guide vanes ahead of the rotor, the rotor both compresses the air and converts rotative energy to pressure energy. With straightener vanes behind the rotor, the rotor both compresses the air and imparts rotative energy to the air.

STATIONARY VANES ACTION

Guide vanes, located immediately ahead of the rotor, impart an initial rotation to the air entering the rotor. This initial air rotation is counter in direction to that of the rotor and is of an amount just sufficient to permit the air to leave the rotor axially.

Straightener vanes, located immediately behind the rotor, partially recover the rotative energy imparted to the air by the rotor and reestablish axial flow to the air leaving the vanes.

Any tendency toward radial air flow due to the rotation imparted to the air by the guide vanes or the fan rotor is negligible for conventional design. This tendency is ignored because the radial pressure gradient of the rotating air mass very nearly provides the centripetal force distribution necessary for radial equilibrium.

MULTI-STAGING

In the multi-staging of vaneaxial fans, it is customary to disregard small air density changes due to compression and to apply identical stages in series. Sometimes, however, the pressure range or compression dictates successive stages of different design. This is frequently the case in design for high fan pressures and always the case for vaneaxial or *turbo* compressor design.

It is theoretically possible to eliminate the use of vanes with a two-stage fan consisting of two differently designed rotors operating in tandem with opposite rotation. The failure of this design to attain popularity is probably due to mechanical complications and commercial aspects.

STACK ACTION

Fan pressure refers to the amount by which the absolute reference pressure (static or total) at the fan outlet exceeds the absolute total pressure at the fan inlet. *Stage pressure* is the amount by which the absolute total pressure at the stage outlet exceeds the absolute total pressure at the stage inlet. Losses between the fan inlet and the stage inlet can be neglected because of accelerated air flow. Therefore, the amount by which stage pressure exceeds fan pressure is referred to as stack loss. The ratio of stack loss to the velocity pressure referred to stage axial air speed is called stack coefficient and is herein designated by the symbol K.

Stack coefficients will vary between the limits of zero and unity, depending upon the efficiency of pressure conversion within the stack, which in turn depends upon the stack design. Crediting a fan with the stage total pressure corresponds to assigning a stack coefficient K equal to zero since losses beyond the stage are excluded. Crediting a fan with the stage static pressure corresponds to a stack coefficient K equal to unity since the fan is thus charged with losses beyond the stage equal to one stage velocity head. There is no need for a stack coefficient to exceed unity because such a stack would be worse than no stack at all.

MINIMUM SPEED FANS

For a particular design capacity, hub size and stack, fan unit speed can be selected within limits by varying blade widths and angles. The upper speed

limit is dictated by efficiency, noise and mechanical considerations, whereas the lower speed limit is reached when further increase of blade width and angle at the blade root or hub fails to provide the required rotor pressure.

With constant axial air speed at all radii it becomes apparent that the blade speed relative to the air must be lowest at the hub where the blade tangential speed is lowest. To compensate for the low blade to air speed at the hub, the blade width must be increased when the available range of blade angle variation alone is inadequate to compensate for the relative speed reduction.

The action of the air passing through the fan rotor is analogous to air passing through a diffuser in that the air speed relative to the rotor blades or diffuser is reduced with an accompanying increase in static pressure in accordance with Bernoulli's theorem. The same is true of the air traversing straightener vanes whereas with guide vanes the air is accelerated at the expense of and with an accompanying reduction of static pressure.

The static pressure increase from the leading to the trailing edge of an airfoil in retarded flow is limited to a fraction of the velocity pressure of the approaching air relative to the airfoil. This fraction or *loading* is approximately 0.63, according to the results of experimentation by Curt Keller[1] with a single symmetrical airfoil in retarded flow at zero degrees attack angle. Above this critical loading air flow, separation occurs at the airfoil surface near the trailing edge and the airfoil *stalls*.

Since vaneaxial fan rotor blades and straightener vanes are actually airfoils in retarded flow, rotor and straightener vane hub loading are influential factors determining minimum design unit speed. Mutual blade and vane effects are more reflected in restriction of maximum airfoil lift than in reduction of lift for any particular attack angle.

DRIVE LOCATION

The performance of vaneaxial fans is impaired to varying degrees by obstructions in the air stream located either just ahead of the fan stage or in the fan stack just beyond the stage. The sensitivity to such obstruction increases with the fan design or operating capacity. High capacity fans are most affected and low capacity least affected.

Obstructions ahead of the fan stage that merely distort the flow pattern of the air entering the stage tend to reduce the stage pressure and may increase the fan power, thereby reducing fan efficiency.

Obstructions ahead of the stage (such as exposed belt drives) that impart rotation to the air in the direction of fan motion reduce the stage pressure and power quite noticeably for small amounts of pre-rotation, but by approximately the same percentage. These obstructions result in little or no effect upon fan efficiency. Fans with straightener vanes are more affected by pre-rotation of the air entering the stage than are fans with guide vanes.

Obstructions of any sort located in the fan stack just beyond the stage have the effect of reducing the stack pressure conversion efficiency without affecting the fan power, thereby reducing the fan efficiency by the same percentage that the fan pressure is reduced.

When vaneaxial fans are driven with belt drives between the fan rotor and

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[1] *Axial Flow Fans*, by Keller-Marks.

the power unit, best results are obtained with drives located ahead of the fan inlet with the belt or ropes covered in the air stream by two streamlined fairings. Thus pre-rotation of the air entering the fan stage is avoided with the minimum possible obstruction to air flow. The fairings are thereby located in low speed air, and the air after passing over the fairings is accelerated upon entering the fan inlet. This acceleration tends to iron out any flow pattern distortion resulting from the obstruction offered by the fairings.

Belt fairings located in the fan stack are in high speed air undergoing deceleration which tends to exaggerate the flow pattern distortion caused by the fairings.

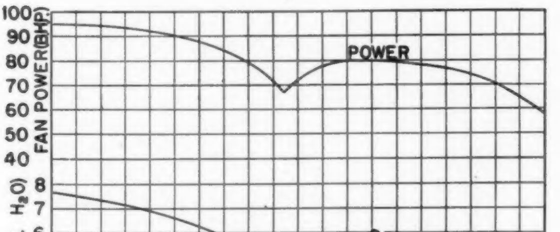

FIG. 2. TYPICAL PRESSURE AND POWER CURVES FOR VANEAXIAL FAN

For High Capacity, Intermediate Speed Fan Equipped with Straightener Vanes

Sometimes it is possible to locate a vaneaxial fan at an angle to the duct approaching or leaving the fan and to extend the drive shaft outside the air stream. This permits the use of short, uncovered belt drives or direct connection to the power unit when speeds allow. Direct connection to the power unit eliminates drive losses. Air duct bends ahead of the fan should be equipped with properly designed turning vanes to direct air into the fan inlet with uniform distribution.

Frequently it is practicable to locate the power unit inside the fan cowling and mount the fan rotor on the shaft of the power unit. Or else separate fan bearings can be provided and the rotor driven by means of suitable coupling to an externally located power unit.

STALL CHARACTERISTICS

Fig. 2 illustrates typical pressure and power curves for a high capacity, intermediate speed vaneaxial fan equipped with straightener vanes. These curves

are for a 60-in. diameter fan at 1168 rpm having a design unit volume of 2050 cfm, design unit speed of 3190 rpm, hub ratio 0.50 and stack coefficient 0.25.

The design volume for this fan at the operating speed is 93,800 cfm which is calculated from the design unit speed and design unit volume obtained by formulas 1 and 2.

The cusps which occur in the two curves at a volume of approximately 47,000 cfm occur for reasons analogous to the reasons that an airplane stalls when attempting to climb too rapidly. This point of operation is therefore referred to as the stall point because at this point the fan blades actually go into an aerodynamic stall like the airplane wing.

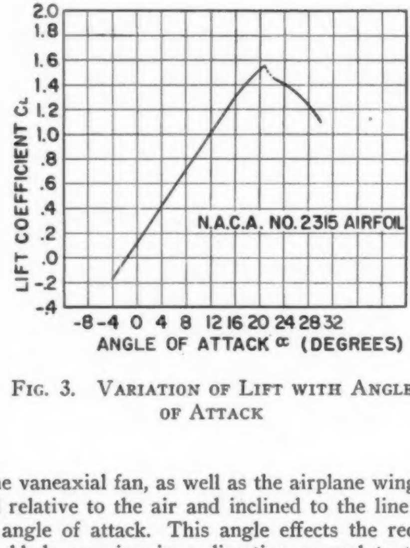

FIG. 3. VARIATION OF LIFT WITH ANGLE OF ATTACK

The blade of the vaneaxial fan, as well as the airplane wing, acts as an airfoil when it is moved relative to the air and inclined to the line of relative motion at the necessary angle of attack. This angle effects the required force or lift exerted upon the blade or wing in a direction normal to the line of relative motion.

Fig. 3 shows how lift varies with angle of attack and is qualitatively typical of fan blade airfoil shapes. It is apparent that there is a value of angle of attack beyond which the lift ceases to increase and falls off precipitously. This maximum angle of attack marks the stall point for the airfoil in question and must not be exceeded for efficient lift.

The angle that the fan blade makes with the plane of rotation is greatly influenced by the ratio of the axial air speed to the tangential blade speed at any particular radius along the blade. This ratio, called discharge coefficient, is minimum at the blade tip where the blade tangential speed is highest and maximum at the blade root or hub where tangential blade speed is lowest for conventional design embodying constant axial air speed at all radii.

The design angle of attack along the fan blade is comparatively constant. It is a small percentage of the total angle between the blade and the plane of

rotation; this total angle in turn is of the same order of magnitude as the angle whose tangent is expressed by the discharge coefficient. Therefore, changes in angle of attack are disproportionate, percentagewise, to corresponding changes in discharge coefficient. This disproportionality increases with increased design discharge coefficient.

Obviously a reduction in discharge coefficient increases angle of attack. Therefore, by reducing the air volume handled by a vaneaxial fan operating at constant speed, the discharge coefficient is reduced at all radii and the angle of

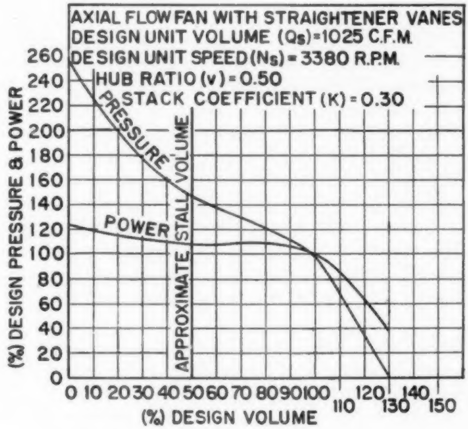

FIG. 4. PRESSURE AND POWER CURVES AND STALL VOLUME FOR VANEAXIAL FAN

attack is increased, especially so at the hub where the discharge coefficient is maximum.

Stalling usually starts at the hub and progresses radially outward along the blade to the tip; this condition is illustrated by the cusps in the pressure and power curves of Fig. 2.

Stalling is physically manifested by pressure instability, change of fan sound to a pulsating roar and severe whirling of the air immediately ahead of the fan rotor. At volumes below the stall point air volume, the pressure becomes steady but the roar and air whirling ahead of the rotor, continue. When operating in this manner, the fan efficiency is very low. However, the fan is not subject to physical harm.

Practically speaking, fans of high capacity or low speed design, or both have a more pronounced stall characteristic than low capacity, high speed fans, because the former types are of high discharge coefficient design.

Fig. 4 illustrates this point with pressure and power curves for an intermediate capacity, intermediate speed family of vaneaxial fans having a design unit volume of 1025 cfm, a design unit speed of 3380 rpm, a hub ratio of 0.50

and a stack coefficient of 0.30. For this fan the stall characteristic is only faintly discernible, manifesting itself by a slight dip in the pressure and power curves.

Considering low first cost and quietness of operation, high-capacity low speed fans are desirable. Fortunately the volume range between the design point and stall point is sufficiently wide to permit application of high capacity, low speed fans to most ventilation systems without likelihood of the fan ever operating in a stall. Even in the case of ventilating systems of variable resistance, in installations such as mines, it is usually possible to so select the fan that stalling at a future date will be avoided.

PRESSURE-VOLUME CURVE

In addition to the fan stall characteristics already discussed, the shape of the fan pressure-volume characteristic curve and influential factors affecting the shape should be considered.

Air volume variation *above* the design volume for a vaneaxial fan operating at constant speed and blade position is accomplished by *decrease* in the air angle of attack along the blade, causing *reduction* in stage pressure. Air volume variation *below* the design volume under the preceding conditions would be accompanied by *increase* in the air angle of attack and would cause *gain* in stage pressure. The degree to which the stage pressure can exceed the design point value by reducing air volume depends upon the margin between the value of the blade air attack angle α at the design point and the maximum or stall value.

The slope dP/dQ of the stage pressure-volume curve ($P =$ pressure, $Q =$ air volume) at design volume is therefore related to the rate of change of attack angle, with volume $d\alpha/dQ$, at the same point. For equal percentage air volume variation, the resulting change in blade air attack angle α is greater for high values of discharge coefficient than for low values, as already explained under the subhead Stall Characteristics.

It follows, therefore, that for a particular design volume and pressure the slope of the stage pressure-volume curve dP/dQ at the design point can be increased or decreased, respectively, (1) given fan diameter and speed—by increasing or decreasing hub diameter; (2) given fan diameter and hub size—by reducing or increasing fan speed; (3) given fan speed and hub diameter—by decreasing or increasing fan diameter.

The effect of pressure losses beyond the stage is to produce a fan pressure-volume curve that is lower in pressure and steeper in slope than the stage pressure-volume curve for all air delivery volumes from the volume of maximum stage pressure to free delivery. The discrepancy in both pressure and slope increases with air delivery volume.

POWER-VOLUME CURVE

A constant speed vaneaxial fan with blades at design pitch or setting has a power-volume curve that can be designed to reach a maximum at the point of design volume or at any other desired operating point within the acceptable operating volume range above the stall region.

Useful air power, PQ, is the product of stage pressure P and air volume Q at any given point of fan operation. By calculus, maximum air power can be shown

to occur at the operating point at which the negative slope dP/dQ of the stage pressure-volume curve is numerically equal to the ratio P/Q.

For practical purposes fan brake horsepower (bhp) can be considered to reach a maximum at the operating point corresponding to maximum air power. Therefore by manipulating fan diameter, speed and hub diameter the fan designer can provide for maximum fan power input to occur at, below, or above the design volume. The fan power-volume curve will, however, be flatter than the

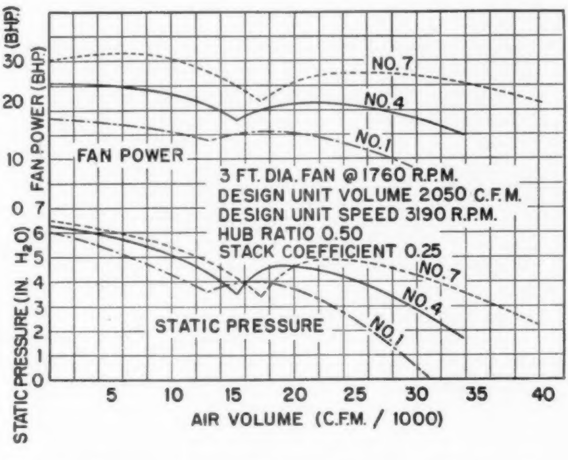

FIG. 5. PRESSURE AND POWER CURVES FOR THREE FAN BLADE POSITIONS

useful air power-volume curve due to the deterioration of stage efficiency for operation above or below design volume.

ADJUSTABLE FAN BLADES

It is possible to adjust the blade setting with vaneaxial fans, thereby altering the fan's characteristics to suit changing pressure-volume characteristics of the ventilation system. Within a limited range of angles the blades continue to operate satisfactorily with stationary unaltered guide or straightener vanes.

Fig. 5 illustrates the point in question by presenting power and pressure curves for a 36-in. diameter vaneaxial fan with straightener vanes having a design unit volume of 2050 cfm, a design unit speed of 3190 rpm, a hub ratio of 0.50 and a stack coefficient of 0.25. The normal or design blade position is the No. 4 position and the fan in question operates at 1760 rpm.

It should be noted that whereas the design volume of this fan is 30,500 cfm, stalling occurs at 13,000, 15,400 and 17,400 cfm for blade positions Nos. 1, 4 and 7 respectively. There is a difference of 6 degrees only in blade angle

between positions 1 and 4 and 4 and 7, which illustrates how sensitive the fan performance is to change in blade position.

The principal purpose of blade adjustment is to maintain high fan efficiency in the face of change in the resistance of the ventilation system. The adjustable feature is not intended for major change in air volume when the fan operates on a system of fixed, unvarying resistance.

Fig. 5 illustrates the previous statement that stall tendency becomes more pronounced with increased discharge coefficient since discharge coefficient is increased with an increase in blade angle.

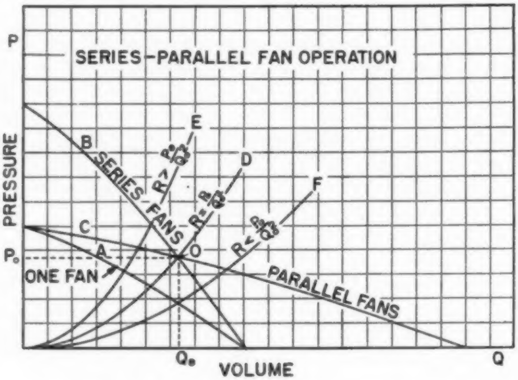

FIG. 6. SERIES-PARALLEL FAN OPERATION

SERIES AND PARALLEL FAN OPERATION

Any two fans, similar or dissimilar, can be operated in series provided that each fan is operated at a speed that will permit passage of the air volume required by the ventilation system.

The parallel operation of two fans requires not only the air volume capacity required by the ventilating system, but each fan must operate at a point on the characteristic fan pressure-volume curve where the slope of the curve is negative. The possibility of one fan blowing air back through the other fan is thereby eliminated, resulting in dependable, steady, parallel operation. Vaneaxial fans are well suited to parallel operation.

The question arises at times as to which type fan operation will produce the highest air volume for a particular ventilation or duct system having a resistance of P/Q^2 units.

Fig. 6 represents two similar fans operating at the same speed. Curve A is the pressure-volume characteristic of each fan. The fans selected are merely of a type having the negative slope pressure-volume characteristic and can be either centrifugal or vaneaxial fans.

Curve B of Fig. 6 is the composite or resultant pressure curve for the two fans operating in series. Curve C is the composite curve for the two fans

operating in parallel. On the pressure-volume curve D for the duct system of resistance P/Q_o^2 units, curves B and C intersect at O, the point at which the two fans will pass the same volume when operated both at the same speed whether operating in series or in parallel.

Duct systems of higher resistance, Curve E, will favor series fan operation. Duct systems of lower resistance, Curve F, will favor parallel fan operation.

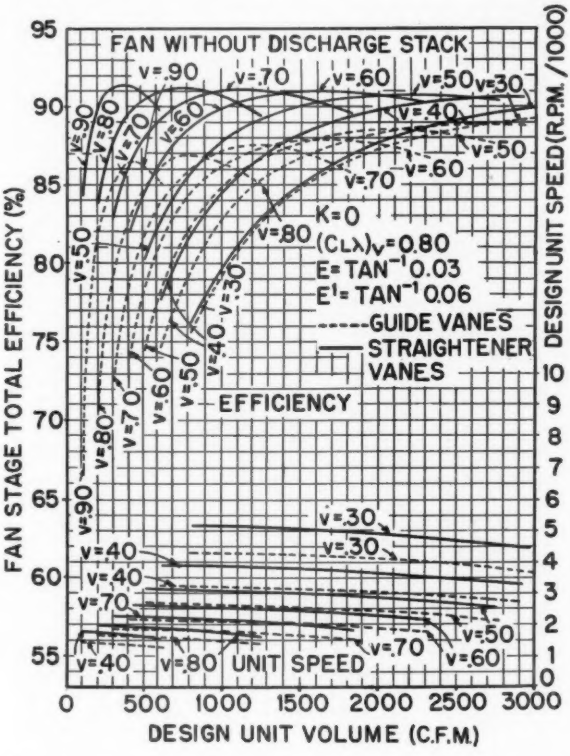

FIG. 7. DESIGN CURVES FOR VANEAXIAL FAN WITHOUT DISCHARGE STACK

This same type analysis is valid for series or parallel operation of dissimilar fans or for similar fans at different speeds or blade positions.

USE OF FAN DESIGN CURVES

The fan design curves, Figs. 7 to 11 inclusive, are based upon a design procedure that strikes a practical balance between fan efficiency, speed and experimentally established permissible hub loadings. These fans are designed to a

constant λ value of 0.80 for the product of blade lift coefficient and ratio of blade width to blade spacing at the blade root or rotor hub. This procedure observes a 0.60 loading limit for the fan rotor, throughout the entire range of fans shown. The 0.60 loading limit is also observed for the straightener vanes for all fans shown except those in the region of extremely low design capacities

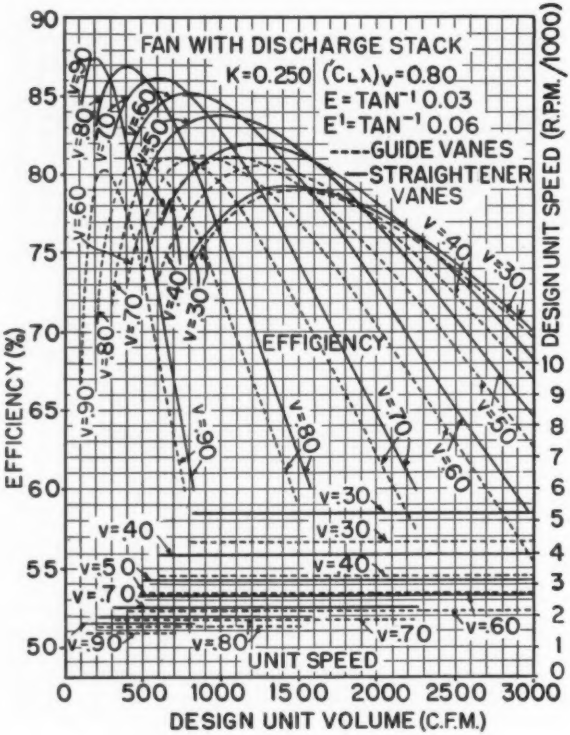

FIG. 8. DESIGN CURVES FOR VANEAXIAL FAN WITH DISCHARGE STACK

where such liberties are permissible because of the relatively low rotational energies in the air entering the straightener vanes.

The aforementioned curves disregard the effects of hub windage and bearing losses. There is no modification for boundary layer effects which are properly negligible for all fans but those with extremely short blades.

An airfoil in an airstream is acted upon by a force called drag in the direction parallel to the relative air flow and a force called lift in the direction normal to the relative air flow. The fan selection curves are based upon as-

sumed airfoil drag to lift ratios of 0.03 and 0.06 for the rotor blades and stationary vanes, respectively.

Fan design curves, Figs. 7 to 11, inclusive for fans with either guide or straightener vanes, plot fan efficiency and design unit speed as ordinates against

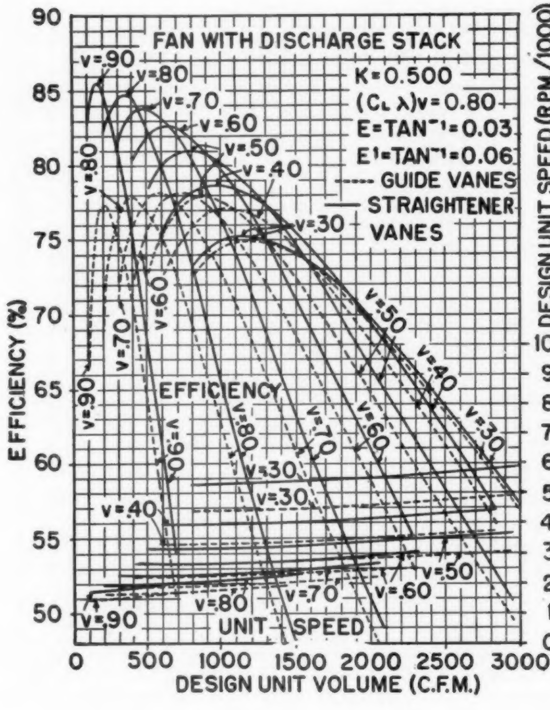

FIG. 9. DESIGN CURVES FOR VANEAXIAL FAN WITH DISCHARGE STACK

design unit volume as abscissae, for different values of stack coefficient *K* and different values of hub ratio.

Examination of the aforementioned curves shows fans with guide vanes to be inherently lower speed fans than fans with straightener vanes. Since guide vanes expand the air and straightener vanes compress the air, the reverse might be expected owing to the greater compression required of the rotor with guide vanes. However, guide vanes impart rotation to the air, counter to the direction of the rotor, thereby maintaining the necessary relative velocities between the air and the fan blade to provide the extra rotor pressure at reduced speed.

These curves further show that high fan efficiency and low blade tip speed, both desirable features, can be incorporated in the same design. Both are best effected with large, slow fans of low design unit volume and high hub ratio.

It is also apparent that for a particular design capacity, Q_s and stack coefficient K, there is a definite value of hub ratio v for maximum fan efficiency.

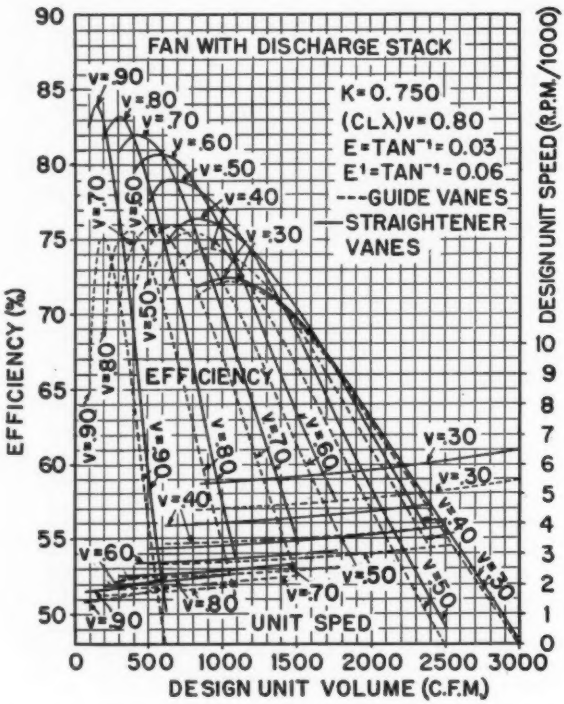

FIG. 10. DESIGN CURVES FOR VANEAXIAL FAN WITH DISCHARGE STACK

Departure above or below this optimum ratio lowers fan efficiency and reduces or increases, respectively, fan speed.

To design a fan of diameter D feet to pass Q cubic feet per minute of air weighing W lb per cubic foot, at a fan pressure P inches water column and a fan speed of N revolutions per minute, the fan design unit volume Q_s and unit speed N_s are calculated by the following formulae respectively:

$$Q_s = \frac{3.65Q}{D^2\sqrt{\frac{P}{W}}} \quad \ldots \ldots \ldots \ldots \ldots \ldots \quad (1)$$

and

$$N_s = \frac{3.65ND}{\sqrt{\frac{P}{W}}} \quad \quad (2)$$

Having thus calculated design unit volume and design unit speed, reference can be made to the fan design curves, Figs. 7 to 11 inclusive, for allowable hub ratios v and approximate fan efficiency for different values of stack coefficient K.

For example, assume a 30 in. diameter fan desired to pass 20,000 cfm of air weighing 0.08 lb per cu ft at a fan pressure of 4 in. water gage and a fan speed of 1750 rpm and a stack coefficient of 0.50 ratio.

First calculate design unit volume with Equation 1 which gives $Q_s = 1650$ cfm. Then calculate design unit speed by Equation 2 which gives $N_s = 2250$ rpm. Reference to the curves of Fig. 9 for 0.50 stack coefficient suggests the design of a fan with guide vanes using a hub ratio of approximately 0.65 with a resulting fan efficiency of approximately 61 percent exclusive of bearing and hub windage losses.

Another example is the design of a 5 ft diameter fan to pass 100,000 cfm at 6 in. water gage with air weighing 0.065 lb per cu ft using a stack with a 0.25 coefficient, having fan efficiency as the sole objective. Using Equation 1, design unit volume is calculated at 1520 cfm. Reference to the curves of Fig. 8 for 0.25 stack coefficient shows the most efficient fan to be a fan with straightener vanes having a hub ratio of approximately 0.50 and a resulting fan efficiency of approximately 81.6 percent exclusive of bearing and hub friction losses. This fan would have a design unit speed of 3140 rpm and would thus operate at a fan speed of 1655 rpm.

The most efficient fan shown with guide vanes would be approximately 80 percent efficient with a 0.40 hub ratio and a design unit speed of 3300 rpm and a fan speed of 1740 rpm. The use of a 0.50 hub ratio with guide vanes would reduce the design unit speed to 2640 rpm and the fan speed to 1390 rpm but would also reduce the fan efficiency to 79 percent which is so slight as to be possibly justified by the marked speed reduction.

It is interesting to note that the use of a 0.60 hub ratio with straightener vanes would result in approximately the same fan efficiency and speed as the use of a 0.50 hub ratio and guide vanes.

It must be remembered that the fan design curves, Figs. 7 to 11, are suggested for convenient reference only and represent but a small percentage of the permissible designs within the 0.60 hub loading limit.

In the last example, for instance, it would be possible to design the 5-ft diameter fan with guide vanes and a 0.60 hub ratio, if desired, to operate at 1390 rpm with a negligible efficiency reduction below that possible with 0.50 hub ratio.

Another example of the use of the curves, Figs. 7 to 11, is the case when given fan diameter, air volume and weight, fan pressure and stack coefficient, it is necessary to select an efficient fan to operate at a limited allowable blade tip speed.

For instance, in the last example suppose the fan blade tip speed had been limited to an upper value of 20,000 fpm. The speed of the 5 ft diameter fan would then be restricted to 1270 rpm. Using Equation 2 gives a corresponding design unit speed $N_s = 2420$ rpm.

Fig. 8 shows at a design unit volume of 1520 cfm and a design unit speed of 2420, it is possible to obtain an approximate fan efficiency of 78.5 percent with guide vanes and 0.53 hub ratio and 77.0 percent with straightener vanes and 0.65 hub ratio. Therefore, guide vanes should be employed for maximum fan efficiency.

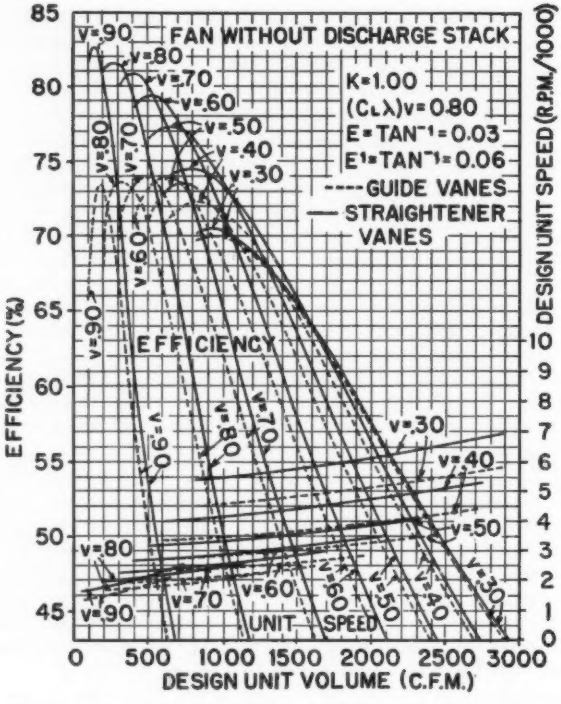

Fig. 11. Design Curves for Vaneaxial Fan without Discharge Stack

The fan design curves will permit rapid determination of the principal fan proportions for any stipulated duty without the necessity of resorting to the customary long and tedious explorative methods involving an intimate knowledge of vaneaxial fan design theory.

Conclusions

The favorable physical and aerodynamic geometry of the vaneaxial fan permits accurate prediction of the air flow pattern throughout the fan and the subsequent utilization of efficient airfoils to compress the air and deliver it to

the system or to a diffuser with uniform flow distribution and for minimum power.

The combination of physical compactness and efficient operation are obtainable to a greater degree with the near axial air path through the vaneaxial fan than with the tortuous air path through the centrifugal fan. The curves of Figs. 7 to 11, illustrate the ability of the vaneaxial fan to combine high capacity and high efficiency.

If quiet operation is required the vaneaxial fan can be designed to meet strict sound level limitations with accompanying further efficiency increase. The aforementioned fan selection curves illustrate the high efficiency potentialities of low capacity, low speed vaneaxial fans with larger hubs.

Some fan applications require duct acoustical treatment with the quietest of fans. In such cases effective duct acoustical treatment behind the fan is more favored by the uniform air flow distribution at the vaneaxial fan outlet than by the uneven outlet air distribution with the centrifugal fan.

The pressure-volume characteristic curve for the vaneaxial can be steep or of gradual slope at the discretion of the fan designer. The fan power-volume characteristic curve can be non-overloading and comparatively flat or can be made to constantly rise or fall throughout the non-stall fan operating range, as desired.

An adjustable bladed vaneaxial fan offers a very wide range of high efficiency fan operation and also permits on-the-job performance alteration without changes in motor or drive. This flexibility is particularly desirable with direct driven fans applied to duct work of unpredictable resistance characteristics.

Vaneaxial fans can be operated in series or in parallel with each other or with centrifugal fans by observing such fundamental requirements as relative capacities and slopes of the pressure-volume characteristic curve of fans to be paralleled.

When handling hot gases, vaneaxial fans can be either belt driven or direct motor driven by ventilating the fan bearings or centrally located motor with fresh cool air conducted to and from the bearings or motor by hollow stationary vanes or properly located pipes.

With the possible exception of conveying certain solids through the fan, there are few types of fan application for which the vaneaxial fan does not present an important alternative to the use of the centrifugal fan.

APPENDIX

FAN DESIGN CURVE CALCULATIONS

The fan efficiency and speed curves, Figs. 7 to 11, are based upon the following assumptions and calculations.

Velocity Diagrams: The velocity diagrams, Figs. A, B, C, and D, are for velocity relationship of air, blades and vanes at some specified radius r for fans with guide and straightener vanes, respectively.

At any radius, the axial component C of the absolute air velocity is considered to be constant for the entire traverse of the air through the fan stage. This assumption disregards the space occupied by the blade and vane profiles, however, close agreement between theory and results appears to justify this assumption.

Stage Coefficient R: The ratio $2p^1/qC^2$ of stage pressure p^1 to the velocity pressure $qC^2/2$ referred to stage axial air speed C is called stage coefficient and will be designated by R.

By definition design unit volume Q_s is the ratio of design volume Q to the product of the square of fan diameter D and the square root of fan pressure P referred to air weighing 0.075 lb per cu ft. If the fan actually handles air of mass density q slugs per cubic foot with a design air volume Q cfm and fan pressure P inches water column, it follows:

$$Q_s = \frac{Q}{D^2\sqrt{\frac{0.075P}{32.17q}}} \quad \text{(A-1)}$$

$$Q = \left(\frac{\pi D^2}{4} - \frac{\pi v^2\ D^2}{4}\right) \cdot 60\ C\ 15\pi D^2\ C\ (1 - v^2) \quad \text{(A-2)}$$

where

v = hub ratio

$$P^1 = 5.2P,\ p^1 = 5.2p \quad \text{(A-3)}$$

$$P^1 = p^1 - \text{stack losses} = p^1 - \frac{KqC^2}{2} \quad \text{(A-4)}$$

$$R = \frac{2p^1}{qC^2} \text{ by definition} \quad \text{(A-5)}$$

From Equations A-3, A-4 and A-5:

$$P = \frac{qC^2}{10.4}(R - K) \quad \text{(A-6)}$$

From Equations A-1, A-2 and A-6:

$$Q_s = \frac{3145\ (1 - v^2)}{\sqrt{R - K}} \quad \text{(A-7)}$$

From Equation A-7:

$$R = \left(\frac{3145\ (1 - v^2)}{Q_s}\right)^2 + K \quad \text{(A-8)}$$

A fan of design unit volume Q_s, hub ratio v and stack coefficient K will operate with a stage coefficient R as calculated with Equation A-8.

Stage Pressure Coefficient ψ: By stage pressure coefficient ψ is meant the ratio $2p^1/qU^2$ of stage pressure p^1 to an air velocity pressure $qU^2/2$ referred to the blade tangential speed U at the radius r in question. Since blade tangential speed U varies directly as the radius r it follows that stage pressure coefficient ψ varies inversely as the square of the radius. Therefore, if at the hub the stage pressure coefficient ψ_o is determined, values at other radii r along the blade are readily computed by the inverse radius squared relationship.

By definition, design unit speed N_s is directly proportional to the product of fan diameter D and fan design speed N and inversely proportional to the square root of fan pressure P referred to air weighing 0.075 lb per cu ft. When handling air of mass density q slugs per cu ft at fan pressure P in. water column, it follows:

$$N_s = \frac{ND}{\sqrt{\frac{0.075P}{32.17q}}} \quad \text{(A-9)}$$

From Equation A-3:

$$P^1 = 5.2P \text{ and } p^1 = 5.2p$$

From Equation A-4:

$$P^1 = p^1 - \frac{KqC^2}{2}$$

From Equations A-3 and A-4:

$$P = p - \frac{KqC^2}{10.4} \qquad \text{(A-10)}$$

$$R = \frac{10.4p}{qC^2} \text{ by definition.} \qquad \text{(A-11)}$$

From Equations A-10 and A-11:

$$P = p\left(\frac{R-K}{R}\right) \qquad \text{(A-12)}$$

$$U_0 = \frac{\pi vDN}{60} \text{ tangential blade speed at hub (ft per sec)} \qquad \text{(A-13)}$$

$$\Psi_0 = \frac{10.4p}{qU_0^2} \text{ by definition} \qquad \text{(A-14)}$$

From Equations A-9, A-12, A-13 and A-14:

$$\Psi_0 = \left(\frac{1275}{v \cdot N_s}\right)^2 \cdot \frac{R}{R-K} \qquad \text{(A-15)}$$

A fan of hub ratio v and stack coefficient K designed to operate at design unit speed N_s and design stage coefficient R will develop a hub stage pressure coefficient ψ_o as calculated by Equation A-15.

When designing a fan of design capacity Q_s and design speed N_s and stack coefficient K, any hub ratio v can be used equal to or greater than the value selected from the curves Figs. 7 to 11. After then calculating stage coefficient R with Equation A-8, stage pressure coefficient ψ_o at the hub can be computed with Equation A-15. Then values of stage pressure coefficient ψ at any desired radii r can be computed from ψ_o by the relation of inverse radii squared.

Stage Discharge Coefficient ϕ*:* By definition stage discharge coefficient ϕ is the ratio C/U of stage axial air speed C to blade tangential speed U at any specified radius r. Since stage pressure coefficient ψ is defined as the ratio $2p^1/qU^2$ it follows that $\psi/\phi^2 = 2p^1/qC^2 = R$. Thus stage discharge coefficient ϕ equals the square root of the ratio of stage pressure coefficient at radius r and stage coefficient R.

$$\Phi = \sqrt{\frac{\Psi}{R}} \qquad \text{(A-16)}$$

After calculating stage coefficient R and pressure coefficient ψ at prescribed radii, values of stage discharge coefficient ϕ at corresponding radii are computed by means of Equation A-16.

Stage Rotation Coefficient m*:* With guide or straightener vanes, the ratio T/U of tangential air speed T imparted by vanes or rotor, respectively, to the blade tangential speed U, at any radius r, is referred to as stage rotation coefficient m.

Rotor Efficiency: At any specified radius r along a rotor blade the compression power dw acquired by the air per differential length of blade dr is the product of the axial air speed C and the axial component of the resultant force dF applied to the air by the blade increment dr (refer to Figs. A and C).

$$dw = dF\cos(B + E) \cdot C \qquad \text{(A-17)}$$

With guide or straightener vanes the rotation-power dw surrendered or acquired respectively by the air to or from the blade increment is the product of half the tangential air speed T and the tangential component of the resultant force dF applied to the air by the blade increment. One half of the tangential air speed T is considered to be the average tangential change in speed experienced by the air while acted upon by the blade increment dr (refer to Figs. A and C).

$$dw = dF \sin (B + E) \cdot T/2 \qquad \text{(A-18)}$$

The total power dw applied to the air by the blade increment is the product of the tangential blade speed U and the tangential component of the resultant force dF applied to the air by the blade increment (refer to Figs. A and C).

$$dw = dF \sin (B + E)\, U \qquad \text{(A-19)}$$

It thus follows that the static or compression aerodynamic rotor efficiency ratio i at a specified radius r is the ratio of the incremental compression air power per blade to the total incremental power applied to the air by the blade increment. Dividing Equation A-17 by A-19:

$$i = \frac{C}{U \tan (B + E)} \qquad \text{(A-20)}$$

With guide vanes this ratio can exceed unity with sufficient air rotation because part of the air compression is at the expense of the air rotation energy.

The total aerodynamic rotor efficiency i at any radius r is the ratio of the algebraic sum of the incremental compression and rotation air power per blade to the total power applied to the air by the blade increment. Equation A-17 plus or minus A-18, divided by A-19:

$$i = \frac{C}{U \tan (B + E)} \pm \frac{T}{2U} \qquad \text{(A-21)}$$

Stationary Vanes Efficiency: At a particular radius r along a guide vane the rotation power dw imparted to the air by the vane increment dr results from the conversion of pressure energy to rotation energy during expansion of the air within the guide vanes. This incremental power conversion is the product of half the tangential air speed T and the tangential component of the resultant force df applied to the air by the vane increment dr (refer to Fig. B).

$$dw = df \cos (\theta + E^1) \cdot T/2 \qquad \text{(A-22)}$$

The total incremental pressure energy decrease at the corresponding vane radius supplies both the acquired incremental rotation power and incremental vane power losses. This expansion power dw is the product of the axial air speed C and the axial component of the resultant force df applied to the air by the vane increment dr (refer to Fig. B).

$$dw = df \sin (\theta + E^1) \cdot C \qquad \text{(A-23)}$$

So the expansion aerodynamic efficiency ratio i at a particular radius r along a guide vane is the ratio of the incremental rotation power imparted to the air per vane to the total incremental expansion air power available per vane. Dividing Equation A-22 by A-23:

$$i = \frac{T \cot (\theta + E^1)}{2C} \qquad \text{(A-24)}$$

At any radius r along a straightener vane the rotation power dw recovered by the vane increment dr is converted to compression power. This incremental power conversion is the product of the axial air speed C and the axial component of the resultant force df applied to the air by the vane increment dr (refer to Fig. D).

$$dw = df \sin(\theta - E^1) \cdot C \quad \ldots \ldots \ldots \ldots \quad (A\text{-}25)$$

At the same radius r along the vane, the total incremental rotation power dw available is the product of half the tangential air speed T and the tangential component of the resultant force df applied to the air by the vane increment dr (refer to Fig. D).

$$dw = df \cos(\theta - E^1) \cdot T/2 \quad \ldots \ldots \ldots \ldots \quad (A\text{-}26)$$

Thus the compressive aerodynamic efficiency ratio i at a specified radius r along a straightener vane is the ratio of the incremental compression air power recovered per vane to the total available incremental rotation air power. Equation A-25 divided by A-26 gives

$$i = \frac{2C \tan(\theta - E^1)}{T} \quad \ldots \ldots \ldots \ldots \quad (A\text{-}27)$$

Stage Efficiency I_s: The total incremental power dw imparted to the air by the fan stage with guide vanes, per blade at any radius r is, the amount by which the incremental blade compression air power exceeds the sum of the incremental blade rotation air power and the air power dissipated within the incremental length of guide vane of efficiency ratio i (refer to Figs. A and B).

$$dw = dF \cdot \cos(B + E) \cdot C - \frac{dF \cdot \sin(B + E)\, T}{2i} \quad \ldots \ldots \quad (A\text{-}28)$$

Substituting in A-28, the value for i from Equation A-24,

$$dw = dF \cdot \cos(B + E) \cdot C - dF \cdot \sin(B + E) \cdot \tan(\theta + E^1) \cdot C \quad \ldots \quad (A\text{-}29)$$

The total incremental power dw imparted to the air by the fan stage with straightener vanes, per blade at any specified radius r is, the sum of the incremental blade compression air power and the portion of the rotation air power recovered by the straightener vanes of efficiency ratio i (refer to Figs. C and D).

$$dw = dF \cdot \cos(B + E) \cdot C + i \cdot dF \cdot \sin(B + E)\, T/2 \quad \ldots \ldots \quad (A\text{-}30)$$

Using Equations A-27 and A-30

$$dw = dF \cdot \cos(B + E) \cdot C + dF \cdot \sin(B + E) \cdot \tan(\theta - E^1) \cdot C \quad \ldots \quad (A\text{-}31)$$

The stage total efficiency ratio i at any radius is the ratio of the incremental total stage air power per blade to the power applied to the air by the blade increment.

With guide vanes ahead of the rotor, or Equation A-29 divided by A-19:

$$i = \frac{C}{U \tan(B + E)} - \tan(\theta + E^1) \frac{C}{U} \quad \ldots \ldots \quad (A\text{-}32)$$

With straightener vanes behind the rotor, dividing Equation A-31 by A-19:

$$i = \frac{C}{U \tan(B + E)} + \tan(\theta - E^1) \frac{C}{U} \quad \ldots \ldots \quad (A\text{-}33)$$

The total incremental stage efficiency can also be derived for fans with either guide vanes or straightener vanes by the following method.

At a given radius r the differential air volume dQ delivered through the stage is the product of the axial air speed C and the differential area of passage $2\pi rdr$. The incremental air power across the stage at the prescribed radius r is therefore the product of the differential air volume dQ and the stage total pressure p^1,

$$dw = C \cdot 2 \cdot \pi \cdot r \cdot dr \cdot p^1 \quad \text{(A-34)}$$

The tangential incremental resistance to the rotor at radius r is the product of the differential air mass per unit time qdQ and the change in tangential speed T imparted to the air. The total power dw applied to the air by the rotor increment is the product of the tangential rotor speed U at radius r and the tangential resistance to the rotor increment,

$$dw = U \cdot q \cdot C \cdot 2\pi \cdot r \cdot dr \cdot T \quad \text{(A-35)}$$

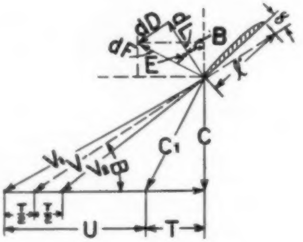

FIG. A. BLADE VELOCITY DIAGRAM

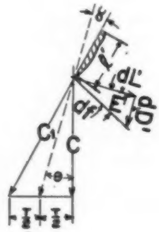

FIG. B. GUIDE VANE VELOCITY DIAGRAM

The stage total efficiency ratio i at any radius r is the ratio of the incremental air-power across the stage to the power applied to the air by the rotor increment. Dividing A-34 by A-35 gives

$$i = \frac{p^1}{qUT} \quad \text{(A-36)}$$

Equations A-32, A-33, and A-36 can be expressed in terms of stage pressure coefficient ψ, stage discharge coefficient ϕ and stage rotational coefficient m as follows:

$$i = \Phi \left[\frac{1}{\tan\left(\tan^{-1} \dfrac{\Phi}{1 + \frac{m}{2}} + E \right)} - \tan\left(\tan^{-1} \frac{m}{2\Phi} + E^1 \right) \right] \quad \text{(A-37)}$$

$$i = \Phi \left[\frac{1}{\tan\left(\tan^{-1} \dfrac{\Phi}{1 - \frac{m}{2}} + E \right)} + \tan\left(\tan^{-1} \frac{m}{2\Phi} - E^1 \right) \right] \quad \text{(A-38)}$$

$$i = \frac{\Psi}{2m} \quad \text{(A-39)}$$

These equations are expressions of the stage efficiency i at any particular radius r. Figs. A, B, C and D show E and E^1 to be the angles made by the resultant differ-

ential forces dF and df and the lift component of these forces, acting between the blade and air and the vane and air respectively, at radius r. The tangents of these angles are the ratios of drag coefficient C_d and lift coefficient C_l for blade and vane, respectively, at radius r. Values of E and E^1 of $\tan^{-1} 0.03$ and $\tan^{-1} 0.06$ are reasonable with sand cast aluminum blades and cast iron vanes coated with some dust and dirt. These values for E and E^1 were used in the preparation of the curves of Figs. 2 to 6 inclusive.

To determine the stage efficiency i at a particular radius r with guide or straightener vanes, respectively Equations A-37 or A-38 and Equation A-39 are equated, after first inserting predetermined values for ψ, ϕ, E and E^1, and m is solved for by the insertion of successive trial values until the equation has been satisfied. Equation A-39 is now used to solve for stage efficiency i at radius r by inserting corresponding values of ψ and m.

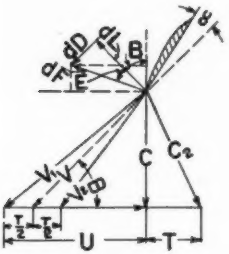

FIG. C. BLADE VELOCITY DIAGRAM

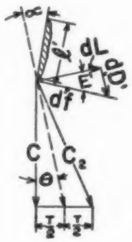

FIG. D. STRAIGHTENER VANE VELOCITY DIAGRAM

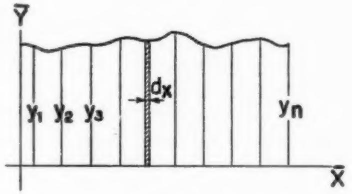

FIG. E. INTEGRATION OF AREA UNDER CURVE

To determine the overall average stage efficiency ratio I_a, the individual values of i at uniformly spaced radii must be first determined and then properly averaged.

The area under any curve such as shown by Fig. E is expressed by:

$$A = \int_{x_0}^{x} y \cdot dx \quad . \ . \ . \ . \ . \ . \ . \ . \ . \ . \ . \ . \ . \quad \text{(A-40)}$$

Such an area can be closely approximated by the Equation

$$A = \left(\frac{y_1 + y_2 - - - - y_n}{n}\right)(x - x_0) \text{ very nearly,} \quad . \ . \ . \ . \ . \ . \quad \text{(A-41)}$$

provided that n is sufficiently large.

Since stage axial air speed C and stage total pressure p^1 are constant at all radii by design, the following formula is an exact expression for the stage air power W.

$$W = \int_{r_0}^{r_1} 2 \cdot \pi \cdot r \cdot dr \cdot C \cdot p^1 \quad . \ . \ . \ . \ . \ . \ . \ . \ . \ . \quad \text{(A-42)}$$

Substituting W for A, $2 \cdot \pi \cdot r \cdot C \cdot p^1$ for y and r for x in Equation A-41 the following formula results

$$W = 2 \cdot \pi \cdot C \cdot p^1 \left(\frac{r_1 + r_2 - - - - r_n}{n}\right)(r_1 - r_0) \text{ very nearly.} \quad . \ . \ . \quad \text{(A-43)}$$

Also the exact expression for the total power input W^1 to the air by the fan rotor is represented by:

$$W^1 = \int_{r_o}^{r_1} 2 \cdot \pi \cdot r/i \cdot dr \cdot C \cdot p^1 \quad \text{(A-44)}$$

wherein i is the stage efficiency ratio at radius r.

Substituting W^1 for A, $\left(2\pi \frac{r}{i} Cp^1\right)$ for y and r for x in Equation A-41 gives the expression:

$$W^1 = 2 \cdot \pi \cdot C \cdot p^1 \left(\frac{\frac{r_1}{i_1} + \frac{r_2}{i_2} \text{----} \frac{r_n}{i_n}}{n} \right)(r_1 - r_o) \text{ very nearly} \quad \text{(A-45)}$$

Average stage efficiency ratio I_s is the ratio of stage air power W to rotor input power W^1, thus Equation A-43 divided by A-45:

$$I_s = \frac{r_1 + r_2 \text{----} r_n}{\frac{r_1}{i_1} + \frac{r_2}{i_2} \text{----} \frac{r_n}{i_n}} \quad \text{(A-46)}$$

very nearly.

Stack Efficiency I_k: Stack efficiency ratio I_k is by definition, the ratio of fan pressure P to stage total pressure p, then

$$I_k = \frac{P}{p} \text{ by definition} \quad \text{(A-47)}$$

From Equation A-12

$$P = p\left(\frac{R-K}{R}\right)$$

Substituting,

$$I_k = \frac{R-K}{R} \quad \text{(A-48)}$$

Thus a fan operating at design specific volume Q_s with a stack coefficient K will operate with a stack efficiency ratio I_k, determined by Equation A-48 after determining stage coefficient R by Equation A-8.

Overall Fan Efficiency I_f: Overall fan efficiency ratio I_f is the product of stage efficiency ratio I_s and stack efficiency ratio I_k, then

$$I_f = I_s \cdot I_k \quad \text{(A-49)}$$

Combining Equations A-46, A-48 and A-49:

$$I_f = \left[\frac{r_1 + r_2 \text{----} r_n}{\frac{r_1}{i_1} + \frac{r_2}{i_2} \text{----} \frac{r_n}{i_n}} \right] \left[\frac{R-K}{R} \right] \quad \text{(A-50)}$$

Equation A-50 is the general expression for overall fan efficiency ratio I_f when operating a fan at design specific volume Q_s and stack coefficient K with incremental stage efficiency ratios i_1, i_2----i_n at radii r_1, r_2----r_n respectively calculated by Equations A-37 or A-38 and A-39, and operating with stage coefficient R as calculated by Equation A-8.

For example suppose it is desired to calculate the efficiency of a 5 ft diameter axial-flow fan with straightener vanes and with a stack having a coefficient $K = 0.25$,

for the duty of passing 100,000 cu ft of air per minute weighing 0.08 lb per cu ft at a fan pressure of 5 in. water column. High efficiency and low speed operation are the chief objectives.

If the fan is to be equipped with sand cast aluminum blades and cast iron straightener vanes, drag/lift ratios of 0.03 and 0.06, respectively, are logical.

With the use of Equation A-1 the fan design unit volume Q_s is calculated as follows:

$$Q_s = \frac{100{,}000}{5^2\sqrt{\dfrac{0.075 \times 5 \times 32.17}{32.17 \times 0.08}}} = 1{,}855 \text{ cfm}$$

Next consult Fig. 8 to obtain the unit speed, hub ratio and approximate efficiency of the lowest speed fan shown which in turn will be approximately the most efficient. The curves recommend a fan with 0.40 hub ratio and design unit speed of 3950 rpm, having an approximate fan efficiency of 79.4 percent, exclusive of hub windage and fan bearing losses. The fan efficiencies shown by Figs. 7 to 11 inclusive are approximate in that they are based upon the calculated efficiency of the stage at the mean radius. The mean radius is the radius of the circle that divides the annulus between the hub and stage casing into two equal area annuli. This practice is sufficiently accurate for purpose of approximation, however, it is now intended to calculate the more exact fan efficiency by considering the stage efficiencies at $0.40r$, $0.55r$, $0.70r$, $0.85r$ and r.

Next calculate stage coefficient R by Equation A-8 as follows:

$$R = \left(\frac{3145\ (1 - 0.40^2)}{1855}\right)^2 + 0.25 = 2.28$$

Then calculate stage pressure coefficient ψ_o existing at the blade root or hub by Equation A-15, thus

$$\Psi_o = \left(\frac{1275}{0.40 \cdot 3950}\right)^2 \cdot \frac{2.28}{2.28 - 0.25} = 0.733$$

Having determined the stage pressure coefficient to be 0.733 at the hub which is at $0.40r$, it is desired to next determine values of ψ at $0.55r$, $0.70r$, $0.85r$ and r. By the rule of inverse square variation these values of ψ become:

$$\Psi = 0.733 \times \left(\frac{0.40r}{0.55r}\right)^2 = 0.389$$

$$\Psi = 0.733 \times \left(\frac{0.40r}{0.70r}\right)^2 = 0.239$$

$$\Psi = 0.733 \times \left(\frac{0.40r}{0.85r}\right)^2 = 0.162$$

$$\Psi = 0.733 \times \left(\frac{0.40r}{r}\right)^2 = 0.117$$

Next Equation A-16 is employed to calculate values of stage discharge coefficient ϕ at $0.40r$, $0.55r$, $0.70r$, $0.85r$ and r as follows:

$$\Phi = \sqrt{\frac{0.733}{2.28}} = 0.568$$

$$\Phi = \sqrt{\frac{0.389}{2.28}} = 0.413$$

$$\Phi = \sqrt{\frac{0.239}{2.28}} = 0.324$$

$$\Phi = \sqrt{\frac{0.162}{2.28}} = 0.268$$

$$\Phi = \sqrt{\frac{0.117}{2.28}} = 0.227$$

It is next necessary to solve for values of stage rotation coefficient m at radii $0.40r$, $0.55r$, $0.70r$, $0.85r$ and r by inserting the calculated values for ψ and ϕ in Equations A-38 and A-39 and successively trying different values for m until equality of the two equations is established. Assume that angles E and E^1 are $\tan^{-1}$ 0.03 and 0.06, respectively, being 1°43′ and 3°26′ respectively.

The following values for m and i result:

r	$0.40r$	$0.55r$	$0.70r$	$0.85r$	$1.00r$
m	0.401	0.214	0.133	0.0917	0.0672
i	0.914	0.907	0.896	0.884	0.871

The average stage efficiency ratio I_s is now calculated with Equation A-46 as follows:

$$I_s = \frac{0.40r + 0.55r + 0.70r + 0.85r + 1.00r}{\frac{0.40r}{0.914} + \frac{0.55r}{0.907} + \frac{0.70r}{0.896} + \frac{0.85r}{0.884} + \frac{1.00r}{0.871}} = 0.889$$

Stack efficiency ratio I_k is now calculated with Equation A-48 as follows:

$$I_k = \frac{2.28 - 0.25}{2.28} = 0.890$$

Finally, overall fan efficiency can be calculated by Equation A-49:

$$I_f = 0.889 \times 0.890 = 0.792$$

The calculated overall fan efficiency of 79.2 percent is in close agreement with the value 79.4 read from the curves of Fig. 8. This agreement is qualitatively correct for all designs covered by the curves of Figs. 7 to 11.

DISCUSSION

W. R. HEATH, Buffalo (WRITTEN): Mr. Mancha's paper presents many facts upon which axial flow designs are based, in a readable and usable form. The method of charting the fundamentals should be of use to a designer if not to a user.

Actually a user could easily be confused or misled by the looseness with which the term "pressure" is used. Fan pressure is defined in the paper as either static or total pressure but several of the references are to "pressure" only and a close study is required to determine whether static pressure or total pressure is intended. Regardless of the relative value to the designer of the two pressures, the fan industry and users of fans in general have for many years used static pressure in rating performance and in figuring system resistances. Most catalog data are clearly so marked. Confusion of the two terms could be especially serious in dealing with vaneaxials or tubeaxials where the energy in the velocity pressure may be as high as 40 percent of the total energy, compared with 10 to 15 percent in centrifugal fans.

The discussion of series and parallel operation and their graphic representation in Fig. 6 is perhaps over-simplified. Duct systems of curve E might favor series operation and duct systems of curve F favor parallel operation as stated in the paper, but only as far as total volume is concerned. Usually the relative horsepowers, the relative noise and the relative stability must also be studied. Again, the ordinate is merely labeled "pressure". With the total volume handled by either the series fans would have twice the velocity and four times the velocity pressure of the parallel fans. Whether or not this velocity energy would be usable depends upon the duct system.

ERNEST SZEKELY, Milwaukee (WRITTEN): I do not have any detailed discussion but it is a very interesting article because vaneaxial fans only recently gained widespread application in the United States and there is, therefore, very little practical data available to the public or even practicing engineers.

It would be extremely interesting if Mr. Mancha submitted some additional data on the subject of parallel fan operation which is illustrated by Fig. 6, but it is only in the nature of a general statement. I do not know whether or not they have comparative data available from tests showing how the performance of a fan is affected when operated in conjunction parallel with another fan of similar size and characteristics.

At a later time, it may be also interesting if Mr. Mancha would write an article on the influence of blade angles and number and size of blades in a practical design.

I hope that Mr. Mancha will pursue this subject which, as I have stated in the beginning, is somewhat new in the American field of fan engineering.

C. W. JOHNSON, Windsor, Ont.: This is an excellent paper from the standpoint of fundamental fan laws being considered and also from the standpoint of design guide with respect to research and development of any particular group of geometrically similar fans. However, this paper does not indicate that performance data given has been verified by actual laboratory tests conducted in accordance with the Fan Test Code of the A.S.H.V.E., and I would like to ask if this is the case.

It is noted with interest that the author, under heading *Drive Location,* states *Air duct bends ahead of the fan should be equipped with properly designed turning vanes to direct air into the fan inlet with uniform distribution.* From a field standpoint this is very important as too often fans as installed have poor entry conditions which do not allow the fan to operate at its design performance.

It is also noted that the author under heading *Stall Characteristics* states that stalling is physically manifested by pressure instability causing a pulsating roar and that this pulsating pressure characteristic does not subject the fan to physical harm. I would like to ask the writer if this condition would not in time accelerate the fatigue of metal in the blades, and if he has had any experience to indicate that this acceleration of fatigue of metal does or does not take place.

Under the heading *Use of Fan Design Curves* it appears that Fig. 7 would be limited to design in research and development as a guide to actual physical tests and would have no practical or field application. This would also apply to Fig. 8 as friction and windage losses are not indicated nor included. Are these design curves to be used for practical application size selection, and if so what correction factors and values of correction factors have to be applied? It is also noted that in the *Use of Fan Design Curves* that fan pressure is not defined as to whether this is static or total pressure in accordance with definitions of same in the Test Code. Is it correct to assume that the author is referring to pressure as total pressure as defined in the Test Code? Also with reference to Fig. 7 is the velocity pressure included in the total pressure that velocity pressure arrived at by considering the velocity in the "NET" area, (*i.e.*, wheel area less hub area)?

With respect to Fig. 8 and the stack coefficient K set at 0.25 would this not require a different design of stack for each different hub ratio thus prohibiting standards of design and making each fan selection a "tailor made" one?

AUTHOR'S CLOSURE: In reply to Mr. Heath; pressure terms and relationships are defined in detail under the heading *Stack Action*. Subsequent pressure reference is to whichever form (static or total) of pressure measurement happens to be under consideration, requiring merely that consistency be observed when comparing fan performances. Under the heading *Series and Parallel Fan Operation* it is clearly stated that Fig. 6 is intended to illustrate comparative air volumes, only, for series and parallel operation of geometrically similar fans. Here again, consistency in pressure reference is all that is required.

In reply to Mr. Szekely; the additional parallel-series fan operational data referred to by Mr. Szekely is contained in the published performance data of each fan manufacturer. The operation of such fans in parallel or series could be easily analyzed by observing the basic fundamentals of parallel-series fan operations. These fundamentals are (*a*) For parallel operation of two fans, total air volume for any particular pressure is equal to the sum of the individual fan air volumes for the same pressure, and (*b*) For series operation, the total pressure for any stated volume is equal to the sum of the individual fan pressures for the same volume.

Another paper dealing with design details might be quite in order and of particular interest to the fan designer, whereas this paper is primarily directed at engineer users of vaneaxial fans.

In reply to Mr. Johnson; the fan design curves are merely intended to illustrate design possibilities, making use of commercially obtainable airfoil shapes and surfaces. Omission of hub windage and journal friction losses amount to from one to 2 percent as reflected by efficiency.

Fan operation in a *stall* for a prolonged period might produce premature failure by fatigue, however, the economics of such unnecessary operation render continued *stall* operation highly improbable. Pressure definition is quite complete under the heading *Stack Action*. I think there are detailed answers under this heading.

Departure from geometrical fan similarity requires dissimilar stacks to obtain constant stack coefficient. However, this departure is limited to variation of stack length only, which is comparatively easy to accomplish.

1394

No. 1394

SEMI-ANNUAL MEETING, 1950

MUSKOKA LAKES, ONT., CANADA

THE 1950 Semi-Annual Meeting of the Society at the Royal Muskoka Hotel, Muskoka Lakes, Canada, provided an opportunity for 333 members and guests to enjoy the hospitality of the Ontario Chapter.

Eleven papers were presented at the three technical sessions held on the mornings of June 19, 20 and 21.

FIRST SESSION, MONDAY, JUNE 19, 9:30 A.M.

Pres. Lester T. Avery, Cleveland, called the Semi-Annual Meeting of 1950 to order at 9:30 a.m. in the Royal Muskoka Hotel. J. H. Fox, general chairman of the Committee on Arrangements of the Ontario Chapter, welcomed the assembled members and guests.

President Avery responded; he emphasized the democratic operation of the Society, and predicted that the Society would continue to make great progress due to the continued cooperation of members in the two great democratic countries, Canada and the United States.

President Avery called upon F. C. Hooper, Toronto, Canada, to present the first paper, Transient Heat Flow Apparatus for the Determination of Thermal Conductivities, by F. C. Hooper and F. R. Lepper (see Chapter 1395).

The second technical paper, Balancing a Steam Heating System by the Use of Orifices, by D. E. Schroeder, Ames, Ia., was presented by the author (see Chapter 1396).

The third technical paper, Plant Research in the Phytotron, by A. J. Hess, Los Angeles, (see Chapter 1397) was presented by the author.

The paper, Thermal Conductivity of Soils for Design of Heat Pump Installations, by G. S. Smith and Thomas Yamauchi, Seattle, Wash. (see Chapter 1398) was presented by title.

The meeting was adjourned at 12:00 noon.

SECOND SESSION, TUESDAY, JUNE 20, 9:30 A.M.

First Vice Pres. L E. Seeley, Durham, N. H., called the meeting to order at 9:30 a.m. aboard the *S.S. Sagamo* which cruised from 11.00 a.m. to 3:30 p.m. through Lake Rosseau and Lake Muskoka.

Dean Seeley called upon G. V. Parmelee, Cleveland, to present the paper, Heat Flow Through Unshaded Glass—Design Data for Use in Load Calculations, by G. V. Parmelee and W. W. Aubele (see Chapter 1399).

Dean Seeley introduced R. H. Heilman, Pittsburgh, who presented the paper, Effective Solar Absorption of Various Colored Paints, by R. H. Heilman and R. W. Ortmiller (see Chapter 1400).

The next paper, A Proposed Psychrometric Chart, by H. B. Nottage, Cleveland, (see Chapter 1401) was presented by the author.

The final paper, A Simple Heated-Thermocouple Anemometer, by H. B. Nottage, (see Chapter 1402) was given in outline by the author.

The meeting was adjourned at 12:15 p.m.

THIRD SESSION, WEDNESDAY, JUNE 21, 10:00 A.M.

Ernest Szekely, Milwaukee, second vice president, called the last technical session to order at 10:00 a.m. in the Rustic Room of the Royal Muskoka Hotel.

The Chairman called on Alfred Koestel, Cleveland, to present the paper, Comparative Study of Ventilating Jets from Various Types of Outlets, by Alfred Koestel, Philip Hermann and G. L. Tuve (see Chapter 1404).

The next paper, Energy Losses in 90-Degree Duct Elbows, by D. W. Locklin, Cleveland, O., (see Chapter 1405) was presented by the author, research engineer of the A.S.H.V.E. Research Laboratory staff.

The final technical paper, Effect of Temperature on Balance of Forced Warm-Air Systems, by N. A. Buckley, S. Konzo, J. M. David and T. L. Towne, all of Urbana, Ill. (see Chapter 1403), was presented by title in the absence of the authors.

Following the presentation of the technical papers and their discussions, Pres. Lester T. Avery assumed the chair and called for a report of the Resolutions Committee which was presented by the chairman, H. L. Stevens.

RESOLUTIONS

WHEREAS, THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS is concluding its 1950 Semi-Annual Meeting in the beautiful Muskoka Lake region of Ontario, Canada; and

WHEREAS, the Ontario Chapter, through J. H. Fox, its president and general chairman of arrangements, and its several committees has extended itself to insure a successful and enjoyable meeting; and

WHEREAS, the Ontario ladies have entertained our wives and families so charmingly; and

WHEREAS, E. Holt Gurney has made us feel again the warm and sincere welcome that Canada has always afforded our Society and has reminded us of the close and cordial relationship between our two great North American countries; and

WHEREAS, the authors of the technical papers and the discussion participants have made valuable scientific contributions to this Society as well as to society at large,

NOW, THEREFORE, BE IT RESOLVED, THAT THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS express by unanimous vote and spread upon the minutes of this meeting a great big thank you to the Officers, the Committees and all of the members of the Ontario Chapter for providing to all those visiting them, the fine program, speakers, publicity, entertainment, transportation and other facilities for enjoyment, and to the authors and the discussors of the technical papers and to our own Society Officers, Committees and staff.

Respectfully submitted,

RESOLUTIONS COMMITTEE, H. L. STEVENS, *Chairman*
H. B. HEDGES, P. J. MARSCHALL

Final announcement of registration for the Semi-Annual Meeting was made as follows: Members 177, ladies 127, guests 29, total 333.

As there was no further business, the chairman declared the Semi-Annual Meeting 1950 adjourned at 11:15 a.m.

HIGHLIGHTS

Entertainment for members and guests was carefully planned by the committee. A ladies' tea was held on Sunday; on succeeding days boat trips, golf tournaments, a party with square dancing and wiener roasts highlighted the meeting.

A Welcome Luncheon, at which E. Holt Gurney, Toronto, honorary chairman of the Committee on Arrangements and past president of A.S.H.V.E. was the speaker, was held on June 19. The Banquet was held on June 20, at which H. D. Henion was the toastmaster, and J. A. Marsh, M.B.E., general manager, *Canadian Exporters Association*, gave an inspiring talk on *This Canada of Ours*.

PROGRAM SEMI-ANNUAL MEETING

Royal Muskoka Hotel, Muskoka Lakes, Ont., Canada—June 19-21, 1950

Saturday—June 17

11:00 a.m.	Committee on Research, R. C. Cross, Chairman (Royal York, Toronto)

Sunday—June 18

1:00 p.m.	REGISTRATION
3:30 p.m.	Council Meeting
4:00 p.m.	Ladies Tea

Monday—June 19

9:00 a.m.	REGISTRATION
9:30 a.m.	TECHNICAL SESSION—Pres. Lester T. Avery presided Welcome Address—John H. Fox, President, Ontario Chapter A.S.H.V.E. Response—Lester T. Avery, President of A.S.H.V.E. Transient Heat Flow Apparatus for the Determination of Thermal Conductivities, by F. C. Hooper and F. R. Lepper Balancing a Steam Heating System by the Use of Orifices, by D. E. Schroeder Plant Research in the Phytotron, by Arthur J. Hess Thermal Conductivity of Soils for Design of Heat Pump Installations, by G. S. Smith and Thomas Yamauchi (presentation by title)
11:00 a.m.	Boat Trip for Ladies (box lunches—return by 3:00 p.m.)
12:15 p.m.	Welcome Luncheon Speaker—E. Holt Gurney, Past President of A.S.H.V.E.
1:30 p.m.	Golf Tournament for Research Cup and Eichberg Memorial Trophy
2:00 p.m.	Joint Meeting of Sub-Committees of TAC on Cooling Load and TAC on Heat Flow through Glass
2:00 p.m.	TAC on the Heat Pump, R. C. Jordan, *Chairman*
2:00 p.m.	TAC on Sensations of Comfort, C. S. Leopold, *Chairman*
2:00 p.m.	TAC on Weather Design Conditions, W. M. Wallace II, *Chairman*
2:00 p.m.	Shuffleboard, Archery, Bowling, Tennis, Horseshoe Pitching, etc.
7:45 p.m.	Chapter Conference Committee, B. L. Evans, *Chairman*
8:00 p.m.	Research Promotion Committee, H. A. Lockhart, *Chairman*
8:00 p.m.	TAC on Insulation, M. W. Keyes, *Chairman*
9:00 p.m.	Muskoka Frolic—Get-together (Square Dancing, Wiener Roast, Beach Party)

Tuesday—June 20

9:00 a.m.	REGISTRATION
9:30 a.m.	TECHNICAL SESSION—First Vice Pres. Lauren E. Seeley presided Heat Flow Through Unshaded Glass, by G. V. Parmelee and W. W. Aubele Effective Solar Absorption of Various Colored Paints, by R. H. Heilman and R. W. Ortmiller A Proposed Psychrometric Chart, by H. B. Nottage A Simple Heated-Thermocouple Anemometer, by H. B. Nottage (presentation by title)

11:00 a.m. Boat Trip (box lunches); also Fast Motor Boat Trip Around Islands (both return by 5:00 p.m.)

2:00 p.m. Nominating Committee, R. T. Kern, *Chairman*

2:00 p.m. Guide Committee, A. B. Algren, *Chairman*

2:00 p.m. TAC on Cooling Load, W. E. Zieber, *Chairman*

2:00 p.m. TAC on Heat Flow Through Glass, R. A. Miller, *Chairman*

2:00 p.m. Informal Golf for Ladies and Gentlemen

2:00 p.m. TAC on Sorbents, John Everetts, Jr., *Chairman*

7:00 p.m. BANQUET—Toastmaster—H. D. Henion

Speaker—John A. Marsh, M.B.E., General Manager, *Canadian Exporters' Association*

Subject—This Canada of Ours

Musical Entertainment and Dancing

Wednesday—June 21

9:30 a.m. REGISTRATION

10:00 a.m. TECHNICAL SESSION—Second Vice Pres. Ernest Szekely presided

Effect of Temperature on Balance of Forced Warm-Air Systems, by N. A. Buckley, S. Konzo, J. M. David, and T. L. Towne (to be presented by R. W. Roose)

Comparative Study of Ventilating Jets from Various Types of Outlets, by Alfred Koestel, Philip Hermann and G. L. Tuve

Energy Losses in 90-Degree Duct Elbows, by D. W. Locklin

Unfinished Business

Resolutions

Adjournment

COMMITTEE ON ARRANGEMENTS

J. H. FOX, *General Chairman*

A. J. STRAIN, *Vice Chairman* E. HOLT GURNEY, *Honorary Chairman*

H. R. ROTH, *Secretary*

Banquet—D. A. Stott, *Chairman;* A. B. Ballagh, D. P. Giffin, J. W. Powlesland.

Entertainment—E. G. Spall, *Chairman;* C. J. Bootes, M. K. Bowman, W. F. Graham, A. E. McGruer, G. E. Pallister.

Finance—H. D. Henion, *Chairman;* D. L. Angus, C. A. Booth, G. R. Mansell, J. H. Ross.

Ladies—W. W. Miller, *Chairman;* L. M. Bennett, John Thompson, Assisted by: Mmes. M. K. Bowman, G. P. Cooper, W. H. Evans, J. H. Fox, N. W. Kingsland, R. H. Lock, G. S. McKernan, W. W. Miller, William Philip, H. R. Roth, E. G. Spall, D. A. Stott, John Thompson.

Publicity—N. W. Kingsland, *Chairman;* K. E. Gould, W. J. Hiscock, James Reith.

Reception—G. P. Cooper, *Chairman;* M. C. Bailey, D. Brock, D. H. Butler, J. W. Darling, K. J. Dewhirst, Robert Dunn, V. J. Jenkinson, J. H. Robinson, E. J. Sandland, H. M. Treleaven, A. W. Wood.

Sessions—William Philip, *Chairman;* Thomas Ferguson, M. W. Shears.

Special Events—G. S. McKernan, *Chairman;* T. R. Barber, A. M. Dion, Arthur Nearingburg, A. G. Ritchie, N. A. Smith.

Sports—R. H. Lock, *Chairman;* Charles Torry, W. J. Usher, Jr.

Transportation—W. H. Evans, *Chairman;* W. J. Coates, Ernest Fox, E. C. Lyons, J. L. Neilans.

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No. 1395

TRANSIENT HEAT FLOW APPARATUS FOR THE DETERMINATION OF THERMAL CONDUCTIVITIES

By F. C. Hooper* and F. R. Lepper**, Toronto, Ont., Canada

History of Project

MARKED limitations to the usefulness of the guarded hot plate apparatus for the determination of thermal conductivity have long been apparent to workers in this field. When work on the thermal conductivity of moist soils was initiated at the University of Toronto four years ago, it soon became apparent that the hot plate apparatus would not yield results having sufficient precision.

The immediate difficulty was two-fold. First, when heat flows due to temperature difference in a moist material, one mechanism of heat transmission is moisture migration with change of phase, due to vapor pressure difference in warmer and cooler positions. The hot plate is a thermal equilibrium device. It will not give a reading until the heat flow has been stabilized, a condition which does not occur until the moisture migration has ceased, at which time the specimen is no longer uniformly wetted, but is dry on the warm side and often more than saturated on the cold face. Thus, neither the moisture migration mechanism effect nor the thermal conductivity in a uniformly wetted specimen exclusive of this mechanism, can be determined.

The second difficulty was the practical problem of obtaining structurally undisturbed soil specimens in the size and shape necessary for the hot plate. This proved exceedingly difficult.

These immediate difficulties, together with the long standing complaints that the hot plate is complicated, expensive, and certainly not portable, that it requires skill in maintenance and operation, and that it will sometimes require days to reach equilibrium and therefore gives rise to delay and high costs in testing programs, led to an investigation of alternative procedures for the determination of thermal conductivity in solids.

The most promising alternative approach appeared to be a transient heat flow procedure utilizing a line heat source. This principle had been adopted by Dr.

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Presented at the Semi-Annual Meeting of The American Society of Heating and Ventilating Engineers, Muskoka, Ontario, Canada, June 1950.

Van der Held[1], who was visited by the first author in 1948, at the University of Utrecht in Holland. Dr. Van der Held developed this principle and applied it to an apparatus for the determination of thermal conductivity in liquids. Earlier references[2, 3] are to be found in the literature, but formerly the theory had not been correctly developed nor properly applied.

From this starting point the object was to develop an engineering scale instrument that would be suited to the testing of building insulations, moist soils in natural location, and other wet or dry solids.

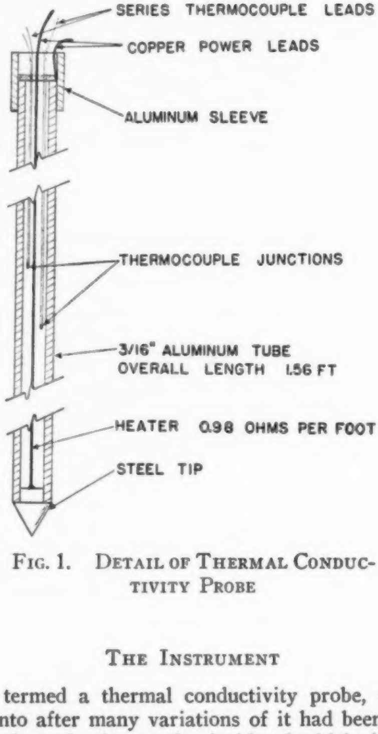

FIG. 1. DETAIL OF THERMAL CONDUCTIVITY PROBE

THE INSTRUMENT

The instrument, termed a thermal conductivity probe, as developed at the University of Toronto after many variations of it had been tested, is shown in Fig. 1. It consists of an aluminum tube, inside of which there is an axial constantan electrical resistance wire, insulated over its length and grounded to the tube at the lower end. An electrical current can be passed through this wire to provide a heat source of constant strength. The lower end of the tube is closed by a pointed plug. The resistor leads are taken out through a seal at the upper end.

Within the tube, and near the center of its length, are located the hot junctions

[1] Exponent numerals refer to Bibliography.

of several thermocouples. These junctions are connected in series with external cold junctions to give temperatures by potentiometer measurements. The thermocouple leads are also taken out through the seal at the upper end of the tube.

Fig. 2 shows diagrammatically, the arrangement of the heater and thermocouples in the measuring circuit.

PRINCIPLE OF OPERATION

The instrument (probe) approximates a line heat source of constant strength in an infinite homogeneous body initially at uniform temperature. The tempera-

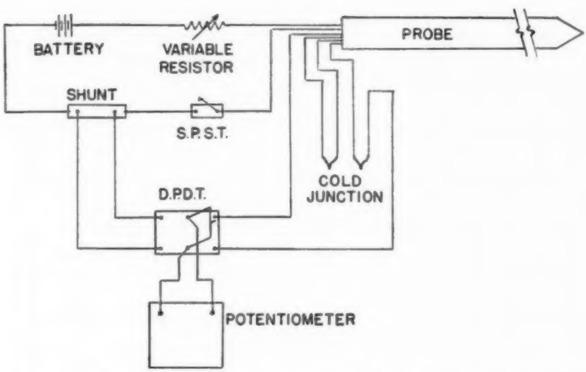

FIG. 2. WIRING AND CONTROL DIAGRAM FOR TEST EQUIPMENT

ture at any point in the body will be a function of several variables including time and thermal conductivity. This relationship is given by the Fourier equation:

$$\frac{\partial\theta}{\partial t} = a\left(\frac{\partial^2\theta}{\partial r^2} + \frac{1}{r}\,\frac{\partial\theta}{\partial r}\right) \qquad (1)$$

where

θ = temperature
t = time
a = thermal diffusivity = $k/\rho C_p$
k = thermal conductivity
C_p = specific heat
ρ = density
r = distance radially from heat source.

This equation can be solved for the particular case of the temperature at the source for a heat input of Q per unit length of heater as outlined in the Appendix. Thus the temperature rise $\Delta\theta$ in the interval between t_1 and t_2 is given by:

$$\Delta\theta = \frac{Q}{4\pi k}\log_e\frac{t_2}{t_1} \qquad (2)$$

where the time t_1 is selected to make terms of a lower order of magnitude in the I function* negligible (Equation 2 is developed as Equation A-14 in the Appendix).

Thus a plot of temperature against $\log_e$ (time) should show a straight portion of slope $Q/4\pi k$. It will be noted that the only property of the body entering this expression is the thermal conductivity, and that the other terms in Equation 2 are quantities readily measurable with the instrument described. This is the basic principle of the method.

Because the probe is of finite diameter, it is necessary to correct the foregoing result for the influence of the instrument itself. A method, due to Van der

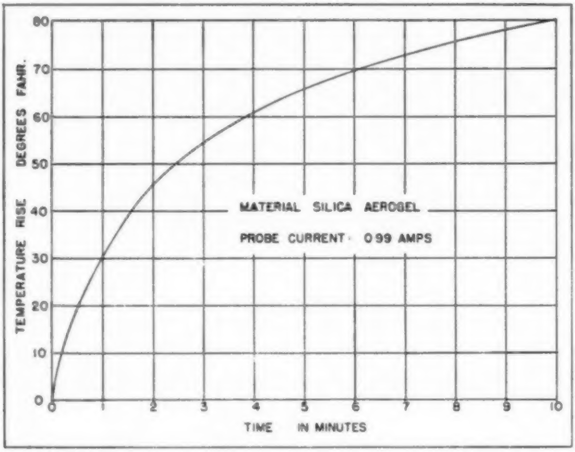

FIG. 3. TIME-TEMPERATURE RELATION IN TEST OF SILICA GEL

Held[1] is available. This method of correction, and the general theory are illustrated by the treatment of the results from an actual test made with an early model of the instrument. The material tested was silica aerogel.

Fig. 3 shows the time-temperature relation as obtained from a recording electronic potentiometer. In Fig. 4 these data are plotted against the logarithm of the time, and the theoretical relation becomes apparent. That is, the data obey the I function* and after a certain time become linear.

To compensate for the finite diameter of the probe, which in effect replaces a small core of the specimen, Van der Held[1] has shown that the difference in heat absorption between the instrument and the displaced core can be considered as a heat production before the start of measured time. That is, a time t_0 is subtracted from each observed time.

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* See Appendix.

By differentiation, Equation 2 will give:

$$\frac{dt}{d\theta} = \frac{4\pi k}{Q} \times t_c \quad \ldots\ldots\ldots\ldots\ldots \quad (3)$$

where

t_c = corrected time. This represents a straight line of slope $4\pi k/Q$, the inverse of the Equation 2 slope.

At $dt/d\theta = 0$, $t_c = t_0$ the time correction.

The data can be plotted on a corrected time scale to give the corrected slope of $Q/4\pi k$. Fig. 5 shows the $dt/d\theta$ *vs.* time relation taken from Fig. 4, indicating

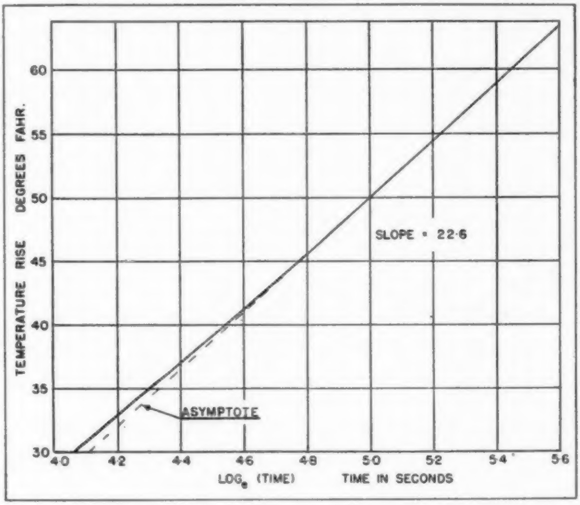

FIG. 4. TEMPERATURE RISE *vs.* LOGARITHM OF TIME

$t_0 = 5$ sec for this probe and material. Fig. 6 shows the data replotted to corrected time, giving the corrected slope 21.2. Note that the necessary condition, that the slopes of Figs. 5 and 6 be reciprocal, is nearly satisfied, slopes being 21.4 and 21.2 respectively. A closer approximation could be made if it were desired.

Thus for this test, substituting actual values

$$k = \frac{Q}{4\pi\, 21.2} = \frac{i^2 R \times 3.42}{4\pi\, 21.2} = 0.0123 \text{ Btu per (hr) (sq ft) (F deg per ft)}$$

where

i = current, amperes.
R = resistance of heater, ohms per foot.

Because, for most materials, t_0 for a particular instrument is very nearly constant, the tedious procedure outlined can be avoided once the value of t_0 for

the instrument has been determined. Moreover, if the time t_1 necessary to reduce the lower order terms of the I function to a negligible quantity is known, only two time-temperature observations need be made. The thermal conductivity is given by:

$$k = \frac{Q}{4\pi\Delta\theta_{(t_2-t_1)}} \log_e\left(\frac{t_2-t_0}{t_1-t_0}\right) = \frac{i^2R\,3.42}{4\pi\Delta\theta_{(t_2-t_1)}} \log_e\left(\frac{t_2-t_0}{t_1-t_0}\right) \quad . \; . \; . \quad (4)$$

The time t_1 can be calculated for any required degree of accuracy, or determined by test as in Fig. 4. For the instrument described in this paper $t_1 = 4$

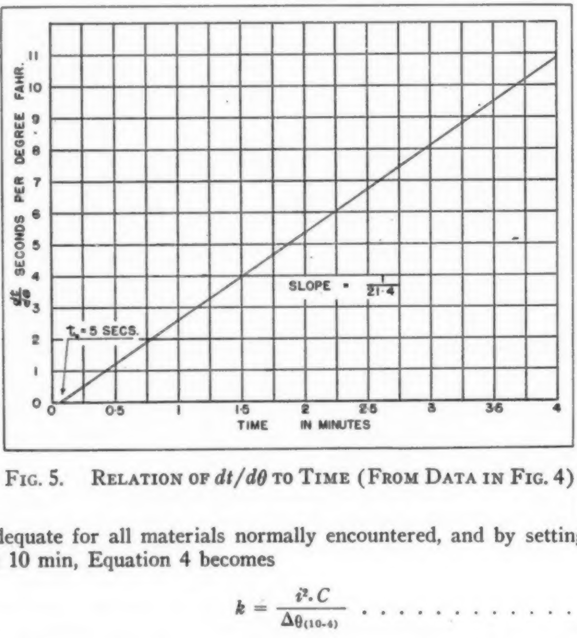

FIG. 5. RELATION OF $dt/d\theta$ TO TIME (FROM DATA IN FIG. 4)

min is adequate for all materials normally encountered, and by setting t_2 arbitrarily at 10 min, Equation 4 becomes

$$k = \frac{i^2 . C}{\Delta\theta_{(10-4)}} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (5)$$

where

$$C = \text{an instrument constant} = \frac{R \times 3.42}{4\pi} \log_e\left(\frac{10 \times 60 - t_0}{4 \times 60 - t_0}\right)$$

$\Delta\theta_{10-4}$ = temperature rise in the interval between 4 and 10 min after start of current.

The calculation then is simply made from Equation 5.

The equivalent mean temperature of the specimen during test can be taken as the initial temperature plus $\Delta\theta_{10-0}/10$ or simply as the initial temperature.

TECHNIQUE OF USING THE PROBE

In reasonably homogeneous and flowing materials such as sand, sawdust and expanded mica, the probe need only be thrust into the specimen so that it is

surrounded on both the side and the ends by at least three inches of the specimen. In hard soils or clays it is permissible to drive a steel rod of smaller diameter than the probe to make a path for the instrument.

With board materials, such as corkboard, a groove to receive the probe must be cut, or a hole be drilled to fit the instrument. Precautions must be taken to avoid air spaces if several boards are laminated to form a specimen.

For gelatins, concretes, waxes and other setting materials the probe can be cast into the specimen. Temperatures must of course be kept below the melting point of the material.

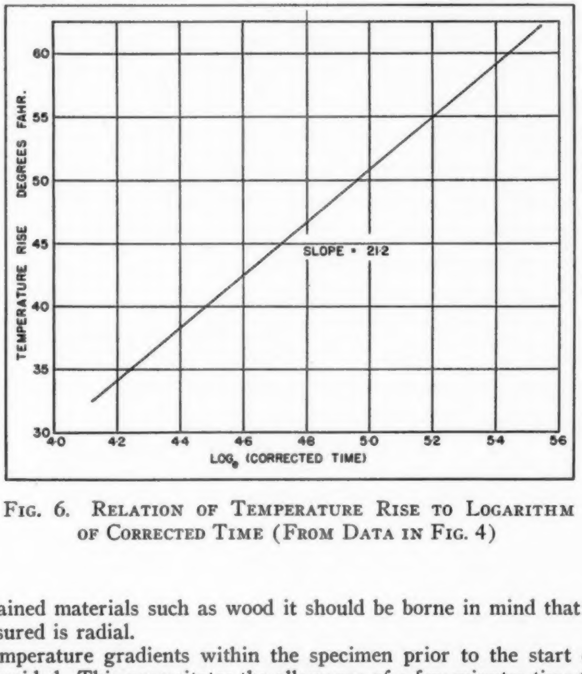

FIG. 6. RELATION OF TEMPERATURE RISE TO LOGARITHM OF CORRECTED TIME (FROM DATA IN FIG. 4)

For grained materials such as wood it should be borne in mind that the heat flow measured is radial.

Any temperature gradients within the specimen prior to the start of a test must be avoided. This necessitates the allowance of a few minutes time to permit the instrument to attain temperature equilibrium with the specimen before a test can be begun. Variation in the temperature of the surroundings of the specimen may also give rise to initial temperature gradients. These can be avoided by wrapping the specimen heavily with insulation some time before a test is to be made. Larger specimens are less subject to this effect.

Time should be kept accurately, and readings of temperature and current should be carried to three significant figures.

Results are calculated directly from Equation 5.

While a wide range of currents and temperature rises can be used, in general lower currents are suited to lower thermal conductivities.

TESTING PROGRAM RESULTS

While only a limited number of tests have so far been conducted with the final form of probe, the instrument has given very good results in tests of both wet and dry soils, sawdust, corkboard, silica gel and similar materials. Fiber insulating materials of very low density have given somewhat erratic results, possibly due to nonhomogeneity in the specimen or to convection currents arising in the interstices. Except in this material no evidence of contact effects has been found.

Results have been reproducible within 0.5 percent. Absolute accuracy is much

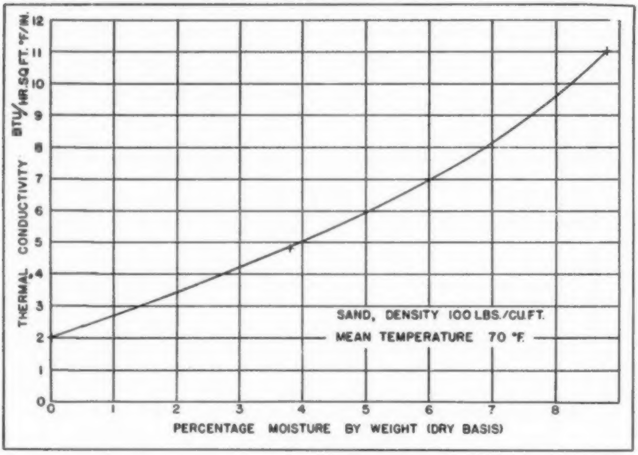

FIG. 7. TYPICAL TEST RESULTS FOR A MOIST MATERIAL

harder to estimate, but it appears to be at least equal to that of the hot plate apparatus for most materials.

The thermal conductivity probe has been very satisfactory in moist materials, particularly in soils. Examinations of the specimens following tests have indicated that no substantial alteration occurs in the moisture distribution during the short test interval. In Fig. 7 a typical set of results for a moist material is shown.

From experience in Toronto in the construction and use of several models of the instrument, several observations with respect to construction can be made:

1. As would be anticipated from the theory, a light probe of high thermal diffusivity, such as one of aluminum, should be used.

2. The thermojunctions, surprisingly enough, appear to be insensitive to radial positioning within the tube, but they must be lacquered, and of fine wire, such as B & S gage No. 33, to avoid time lag. The use of three or more thermocouples in series not only increases the precision of temperature measurement but avoids effects due to

small local anomalies in the specimen. All thermojunctions should be near the center of the tube length. A spacing of about ¾ in. along the tube axis seems satisfactory.

3. The use of constantan for the resistance wire has the obvious advantage of constant current for constant voltage when temperature varies. Attempts to use a heater wire having a temperature coefficient with a view to determining temperature by the resistance variation not only lead to complicated controls to yield constant power but introduce end effect errors as well.

4. The size of the instrument can be varied widely, but a length to diameter ratio of less than 100 seems inadvisable. Large instruments require very large specimens, and small instruments are subject to slight anomalies in the specimen. For general utility, reasonable mechanical strength, and satisfactory use in coarse material, the instrument shown in Fig. 1 has been satisfactory.

5. The use of direct current avoids induction effects, permits current measurement by potentiometer, and makes it possible to use a battery and so make the instrument portable.

PRESENT STATUS OF PROJECT

The initial stage in the research program was primarily directed toward the development of a satisfactory method for the determination of thermal conductivity in moist soils. The instrument developed for this purpose has been satisfactory, and it has been successfully applied to other materials, such as building materials and insulation.

The following advantages are associated with the new method:

It will measure with equal precision the thermal conductivity of wet or of dry materials.

It will permit the testing of materials such as soils, in natural location, and consequently structurally undisturbed.

The apparatus is compact and portable. It is relatively inexpensive to construct and operate.

A determination can be made in a few minutes, and the operator requires little skill or training. The required calculations are very simple. The method is suited to production control application.

The degree of precision appears adequate for engineering use.

It should be recognized that the instrument is a primary standard apparatus. The development of the equations involved the introduction of no arbitrary constants. The instrument does not require calibration against any other standard or material.

The project is being extended at the University of Toronto in two directions: (1) more data are being accumulated both in the field and in the laboratory on the thermal conductivities of various materials, including rocks and frozen soils and (2) a smaller instrument is being developed for use in the direct determination of thermal conductivity of meats, vegetables and other foodstuffs, particularly those normally stored by cooling or freezing.

ACKNOWLEDGMENT

The work has been carried out in the Heat Transfer Laboratory at the University of Toronto. The authors gratefully acknowledge the financial assistance obtained from the *National Research Council* at Ottawa.

APPENDIX

The heat flow from a line source in an infinite specimen is given by the Fourier equation:

$$\frac{\partial\theta}{\partial t} = a\left(\frac{\partial^2\theta}{\partial r^2} + \frac{1}{r}\frac{\partial\theta}{\partial r}\right) \quad \text{(A-1)}$$

where

θ = temperature.
a = the thermal diffusivity of the specimen.
t = time.
r = the radius from the source.

A solution of this partial differential equation may be developed in the following manner[5].

Consider unidirectional heat flow in the direction x subject to the Fourier equation:

$$\frac{\partial\theta}{\partial t} = a\frac{\partial^2\theta}{\partial x^2} \quad \text{(A-2)}$$

with the restricting condition that $\theta = f(x)$ when $t = 0$.

Substitute in Equation A-2 $\theta = e^{bt+cx}$ where b and c are parameters, giving $b = ac^2$.

Now setting $c = \pm i\gamma$

$$\theta = Le^{-a\gamma^2 t}\ e^{i\gamma x}$$

or

$$\theta = Me^{-a\gamma^2 t}\ e^{-i\gamma x}$$

But since

$$e^{\pm i\gamma x} = \cos\gamma x \pm i\sin\gamma x$$

combining the two expressions for θ by addition and choosing suitable values of L and M gives

$$\theta = e^{-a\gamma^2 t}\cos\gamma x$$

and

$$\theta = e^{-a\gamma^2 t}\sin\gamma x$$

which are particular solutions of Equation A-2 for any value of γ. Multiply these by B and C, any functions of γ, and obtain the sum of the infinite series given by:

$$\theta = \int_0^\infty (B\cos\gamma x + C\sin\gamma x)\, e^{-a\gamma^2 t}\, d\gamma \quad \text{(A-3)}$$

which is also a solution of Equation A-2.

Determine functions B and C such that, for $t = 0$, Equation A-3 becomes $= f(x)$ in accordance with the restriction.

Fourier's Integral gives

$$f(x) = \frac{1}{\pi}\int_0^\infty \int_{-\infty}^\infty f(\lambda)\cos\gamma(\lambda - x)\, d\lambda$$

and from Equation A-3 when $t = 0$ this must equal

$$\int_0^\infty (B \cos \gamma x + C \sin \gamma x)\, d\gamma$$

Therefore

$$B = \frac{1}{\pi} \int_{-\infty}^{\infty} f(\lambda) \cos \gamma \lambda\, d\lambda$$

and

$$C = \frac{1}{\pi} \int_{-\infty}^{\infty} f(\lambda) \sin \gamma \lambda\, d\lambda$$

Substituting in Equation A-3 gives

$$\theta = \frac{1}{\pi} \int_0^\infty e^{-a\gamma^2 t}\, d\gamma \int_{-\infty}^{\infty} f(\lambda) \cos \gamma (\lambda - x)\, d\lambda$$

which is the required solution of θ for any values of x and t.

But since

$$\int_0^\infty e^{-m^2 y^2} \cos ny\, dy = \frac{\sqrt{\pi}}{2m} e^{-\frac{n^2}{4m^2}}$$

where m and n are arbitrary constants. It follows that:

$$\theta = \frac{1}{\sqrt{4\, at\, \pi}} \int_{-\infty}^{\infty} f(\lambda)\, e^{-\frac{(\lambda - x)^2}{4at}}\, d\lambda \quad \text{(A-4)}$$

A special form of the previous case is obtained if the surroundings have an initial temperature of 0. Consider that for each positive temperature at distance $+x$ there is an equal negative temperature at distance $-x$. To express this mathematically, suppose that for points on the positive side of the origin $\lambda = \lambda_1$ and on the negative side $\lambda = \lambda_2$. Then λ_1 and λ_2 are positive and the temperature $f(\lambda)$ can be expressed as $f(\lambda_1)$ for the positive region and $-f(\lambda_2)$ for the negative region. For this special case Equation A-4 can be written

$\left(\text{substituting } n = \frac{1}{2\sqrt{a\,t}}\right)$:

$$\theta = \frac{n}{\sqrt{\pi}} \left[\int_0^\infty f(\lambda_1)\, e^{-(\lambda_1 - x_1)^2 n^2}\, d\lambda_1 + \int_\infty^0 -f(\lambda_2)\, e^{-(-\lambda_2 - x)^2 n^2} (-d\lambda_2) \right]. \quad \text{(A-5)}$$

If, instead of assuming the unidirectional flow of Equation A-2 one considers radial heat flow in a plane from a point, the differential equation becomes

$$\frac{\partial \theta}{\partial t} = \frac{a}{r} \frac{\partial^2 \theta r}{\partial r^2}$$

or

$$\frac{\partial (r\theta)}{\partial t} = a \frac{\partial^2 (r\theta)}{\partial r^2} \quad \text{(A-6)}$$

with the restriction that

$$\theta = f(r) \text{ when } t = 0$$

Let $\mu = r\theta$ and Equation A-6 reduces to

$$\frac{\partial \mu}{\partial t} = a \frac{\partial^2 \mu}{\partial r^2} \quad \ldots\ldots\ldots\ldots\ldots \quad \text{(A-7)}$$

for which a solution is given by Equation A-5. Using λ as the variable of integration and remembering that $\mu = \lambda f(\lambda)$ for $t = 0$, the temperature at distance r from the point is given by

$$\mu = r\theta = \frac{n}{\sqrt{\pi}}\left[\int_0^\infty \lambda f(\lambda)\, e^{-(\lambda-r)^2n^2}\, d\lambda - \int_0^\infty \lambda f(\lambda)\, e^{-(\lambda+r)^2n^2}\, d\lambda\right] . \quad \text{(A-8)}$$

If the initial temperature is constant, at θ_o within a small sphere of radius R and is 0 everywhere outside in the infinite solid, from Equation A-8

$$\theta = \frac{\theta_o}{r}\frac{n}{\sqrt{\pi}}\left[\int_0^R \lambda e^{-(\lambda-r)^2n^2}\, d\lambda - \int_0^R \lambda e^{-(\lambda+r)^2n^2}\, d\lambda\right] \quad . \; . \quad \text{(A-9)}$$

To reduce this finite sphere to a point source let the radius of the sphere become vanishingly small. Also, put $Q = \theta_o C_p\rho(4/3)\pi R^3$ as the heat in the sphere, and substitute θ_o in Equation A-9.

$$\theta = \frac{3Qn}{4C_p\rho R^3\pi^{3/2}r}\left[\int_0^R \lambda e^{-(\lambda-r)^2n^2}\, d\lambda - \int_0^R \lambda e^{-(\lambda+r)^2n^2}\, d\lambda\right] \quad . \; . \; . \quad \text{(A-10)}$$

Now

$$e^{-(\lambda-r)^2n^2} = e^{-\lambda^2n^2}\, e^{2\lambda n^2 r}\, e^{-r^2n^2}$$

but

$$e^y = 1 + y + \frac{y^2}{2!} + \ldots\ldots.$$

So

$$e^{-(\lambda-r)^2n^2} = \left(1 - \lambda^2n^2 + \frac{\lambda^4n^4}{2!} + \ldots..\right)\left(1 + 2\lambda rn^2 + \ldots..\right) e^{-r^2n^2} \quad . \; . \; . \quad \text{(A-11)}$$

But since $0 < \lambda < R$, λ is very small and Equation A-11 becomes

$$(1 + 2\lambda rn^2)\, e^{-r^2n^2}$$

Then from Equation A-10

$$\theta = \frac{3Qn}{4C_p\rho R^3\pi^{3/2}r}\, e^{-r^2n^2}\left[\int_0^R \lambda(1 + 2\lambda rn^2)\, d\lambda - \int_0^R \lambda(1 - 2\lambda rn^2)\, d\lambda\right]$$

$$= \frac{3Qn}{4C_p\rho R^3\pi^{3/2}r}\, e^{-r^2n^2}\left[\frac{4R^3rn^2}{3}\right] = \frac{Qa}{k}\left(\frac{n}{\sqrt{\pi}}\right)^3 e^{-r^2n^2} \quad . \; . \; . \quad \text{(A-12)}$$

which is the equation for the temperature distribution about an instantaneous point source. If a series of these sources be combined in a straight line an instantaneous

line source will result. The magnitude of each source will be $Q\,d\,z$ where Q is the heat released per unit length of z. Thus summing the effect of these point sources,

$$\theta = \int_{-\infty}^{\infty} \frac{Qa}{k}\left(\frac{n}{\sqrt{\pi}}\right)^3 e^{-(r^2+z^2)n^2} dz = \frac{Qa}{k}\left(\frac{n}{\sqrt{\pi}}\right)^3 e^{-r^2n^2} \int_{-\infty}^{\infty} e^{-z^2n^2} dz$$

or

$$\theta = \frac{Qan^2e^{-r^2n^2}}{k\pi} \quad \ldots\ldots\ldots\ldots\ldots \quad \text{(A-13)}$$

as the temperature distribution around an instantaneous line source.

A continuous line source results upon integration of Equation A-13 in

$$\theta = \frac{Q}{k4\pi}\int_0^t \frac{e^{-\frac{r^2}{4at}}}{t}\,dt$$

Substituting

$$\beta = \frac{r}{2\sqrt{at}}$$

one obtains

$$\theta = \frac{Q}{k2\pi}\int_{rn}^{\infty} \frac{e^{-\beta^2}d\beta}{\beta} = \frac{Q}{2\pi k}\,[\,I(rn)\,]$$

where the series I is given by

$$I(rn) = C - \log_e(rn) + \frac{(rn)^2}{2} - \frac{(rn)^4}{8} + \ldots\ldots$$

and if $r\,n$ is sufficiently small

$$I(rn) = C - \log_e(rn)$$

or

$$\theta = \frac{Q}{2\pi k}\,[C - \log_e(rn)]$$

Between times t_1 and t_2 the temperature rise is given by

$$\Delta\theta = \frac{-Q}{2\pi k}\,[\,\log_e(rn_2) - \log_e(rn_1)\,] = \frac{-Q}{2\pi k}\log_e\left(\frac{n_2}{n_1}\right)$$

or

$$\Delta\theta = \frac{Q}{4\pi k}\log_e\left(\frac{t_2}{t_1}\right) \quad \ldots\ldots\ldots\ldots \quad \text{(A-14)}$$

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DISCUSSION

L. R. INGERSOLL, Madison, Wis.: I am very glad to see the line source equation again prove its usefulness, but I must confess that, before reading this paper, I would have questioned whether the described scheme would work as well as it does. I would have feared radial temperature gradients within the tube and would have expected that the neglect of the radius r in Equation A-14 would introduce errors in some soils. I would also have raised still other objections.

However the fact that the scheme works wipes out most of my objections. My guess is that it will eventually be found in error for absolute determinations and it may also require some correction factors when used with materials of greatly different conductivity and diffusivity, but as a quick-action comparison method for the rapid conductivity assay of soils it should prove exceedingly useful.

E. R. QUEER, State College, Pa.: This instrument indeed is a nice development and will be quite convenient. There are, however, several questions that I would like to raise, one being at what temperature the probe is operated? That is rather important because if you elevate the temperature over the normal ambient temperature, where you are injecting it into the material, you start moisture migration as soon as the temperature is elevated. Perhaps the migration may not be large but it will have a transient effect.

Secondly, the hot plate, admittedly is an instrument that can be used only on dry materials; however, it has been an exceedingly accurate instrument. Albeit a bit cumbersome, it is still a very fine instrument to be used in thermal conductivity work.

R. S. DILL, Washington, D. C.: The authors are to be congratulated on this development. I do not see why, at the moment, moisture migration is not a problem with this instrument the same as it is with the hot plate. Also, it appears that the indicated conductivity of a soil would be an instantaneous effect and that the real conductivity would be more nearly indicated by a continuous measurement.

AUTHOR'S CLOSURE: The questions of both Professor Queer and of Mr. Dill, with respect to moisture migration, reveal the inadequacy of our description of moisture migration mechanism in heat flow in solids, and the relationship of this mechanism to the probe readings.

Moisture migration will always take place when a temperature difference exists in a permeable moist material. In most cases this effect is one of evaporation in the warm position, transmission of the vapor by diffusion to the cooler position, and condensation in the cooler position. Heat, the larger part in the form of latent heat, is thus transmitted by this mechanism. This is in addition to the conducted heat.

Except under special conditions, heat flow in a moist solid is always a transient or non-steady flow, since the migration of the moisture is continuously altering the condition and properties of the material. It is therefore necessary to determine an instantaneous thermal conductivity which includes the effect of moisture migration corresponding to the particular moisture content. With the probe method, this is accomplished by making a measurement in an initially uniformly moistened specimen in a time period and with a temperature rise which does not permit large alterations in properties of the material to take place. For example, in a six minute test with a 30 deg temperature rise, the moisture content of the material within ¼ in. of the probe changed from 10.0 to 9.8 percent. Thus, the thermal conductivity determined

in this test is the instantaneous value for heat flow in the material at 10 percent moisture content.

The application of the results to, let us say, heat flow from a buried pipe in moist soil, involves not only a knowledge of the initial soil moisture, but a knowledge of the moisture distribution at the time of interest. Even knowing this distribution,

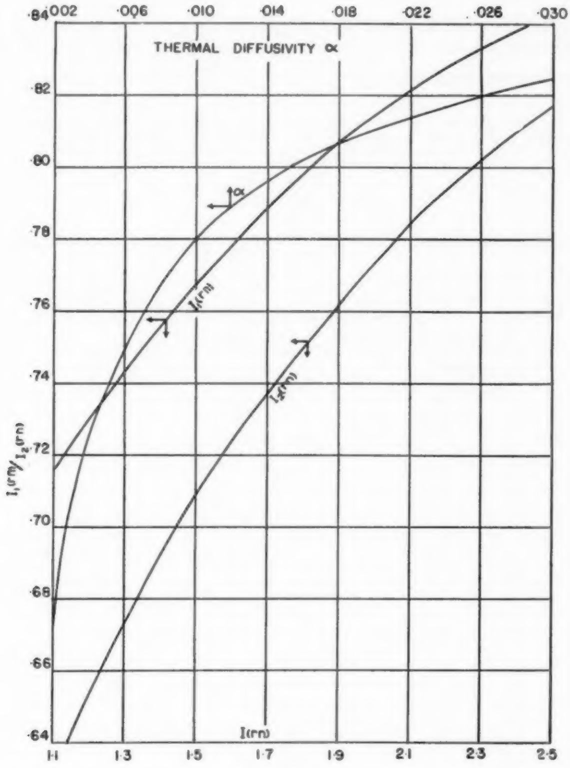

FIG. A. RATIO OF THE *I(rn)* FUNCTION AT TWO CHOSEN TIMES PLOTTED AGAINST THERMAL DIFFUSIVITY

the heat flow calculation is exceedingly complicated, since it involves flow through layers of successively altering thermal properties.

I agree with Professor Queer that the hot plate is a very accurate instrument when properly used and with dry materials. Unfortunately, it is not always used within these limitations.

Professor Ingersoll's comments are of remarkable accuracy. Additional experience with the probe has indicated that a higher degree of absolute accuracy is obtainable when the thermal capacity of the probe is matched to the thermal capacity of the

specimen. This implies heavier walled tubes for testing more dense materials. An investigation of this and associated effects is being made.

Following a suggestion by Dr. Misener of the University of Western Ontario, we find it possible to obtain values of the thermal diffusivity and of the specific heat from the probe readings. Because very little work has as yet been possible on this aspect of the project, no claim to accuracy or utility is made, but the method is outlined here for those who may find it of interest.

The values of the $I(rn)$ function, and of the ratio of the $I(rn)$ function at two chosen times can be obtained and plotted for a range of values of thermal diffusivity α, as in Fig. A. Because the observed ratio $\frac{\Delta\theta_{4-0}}{\Delta\theta_{10-0}} = \frac{I_1(rn)}{I_2(rn)}$, α can be obtained directly from the curve.

At this α also, one can evaluate $I_1(rn)$ and $I_2(rn)$, and obtain k from the relation

$$k = \frac{Q}{2\pi\,\Delta\theta} \cdot I(rn).$$

For the results of Fig. 3 in the paper, $\frac{\Delta\theta_{4-0}}{\Delta\theta_{10-0}} = \frac{61}{80} = 0.763$ gives $\alpha = 0.0073$ and $k = \frac{Q}{2\pi\,\Delta\theta} \cdot I(rn) = \frac{3.20}{2\pi\; 61}\; 1.465 = 0.0123$ at four minutes and $k = 0.0124$ at 10 minutes. This checks well with tabular values and with the results obtained from the other method of calculation for this particular material.

Since density is easily determined, this permits direct calculation of the specific heat as $C_p = \frac{k}{\rho\;\alpha} = \frac{0.0123}{8.5 \times 0.0073} = 0.198.$

1396

No. 1396

BALANCING A STEAM HEATING SYSTEM BY THE USE OF ORIFICES

By D. E. Schroeder*, Ames, Ia.

THE VARIOUS steps to be taken in obtaining proper steam distribution by means of orifice plates installed in the valves of a steam heating system are illustrated in the description of the installation which was made in the Service Building of Iowa State College, Ames, Ia.

After two years of unsatisfactory operation, a successful change was effected in this building through the installation of a drilled orifice plate installed in each radiator valve between the valve body and tail piece.

Because the troubles encountered in this building are not unique in the heating industry, and because records were kept of the steam consumption for the building, the information and experience gained from this job should prove valuable to heating engineers and building owners who may be considering the installation of orifices as a means of correcting steam distribution difficulties.

The Service Building is only one of many buildings on the campus having orifice plates as a means of controlling steam distribution.

The Service Building (Fig. 1) is a T-shaped, three-story building, without basement, of brick and tile construction. The rectangular main part of the building, having a full north exposure, is 149 ft 9 in. long × 44 ft 9 in. wide and has extending from the center on the south side a 62 ft 9 in. × 29 ft annex. The visual instruction service, printing department, radio station and various other service agencies are housed in the building which was completed in 1939. With the exception of the broadcasting studios and a film storage vault, the entire building is heated by direct radiation. The broadcasting studios, located on the top floor, have controlled winter and summer temperature and humidity regulated by the condition of the air used for ventilation. The film storage vault, located on the ground floor, is completely surrounded by warm areas and requires cooling the year around.

Similarity of occupancy and usage plus the uniform building construction and the consistent use of the same type of radiation throughout the building made this heating installation highly adaptable to zoning and zone control. The shape of the building and its physical arrangement dictated that the building be divided

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Presented at the Semi-Annual Meeting of The American Society of Heating and Ventilating Engineers, Muskoka, Ontario, Canada, June 1950.

into three heating zones including the rooms having north, southeast, and southwest exposures, respectively.

Wall-hung tubular radiators supplied the heat for the rooms and halls. Steam at 3 psig was supplied to the direct radiation through up-feed risers from zone mains located in an underground pipe trench around the periphery of the building. The steam for the building was supplied by the college heating plant and taken from the distribution mains in the tunnel at a point near the northwest corner of the building. All of the condensate was collected in a single pipe and was metered before being returned to the heating plant.

FIG. 1. VIEW OF NORTH SIDE OF SERVICE BUILDING, IOWA STATE COLLEGE, AMES, IA.

The essential control elements in each zone consisted of a control responsive to weather conditions and mounted on the outside of the building in a location representative of the exposure for the zone, plus an automatic valve, located ahead of the first radiator in the zone. The valve was maintained in either a fully open or fully closed position by the outside control. The three zones were controlled in an identical manner. Night and holiday shut downs were accomplished by a program clock in the control circuit set to switch the control, at a predetermined time, to a low-limit thermostat installed in the building. In the morning the control function was automatically returned to the outside control. An adjustable type orifice plate was inserted between each radiator valve and valve tail piece.

The winter of 1939-40 was the first season of operation for the system. A considerable amount of general service work was required to remove defects in the mechanical parts of the system. Numerous complaints concerning uncom-

fortable room temperatures were reported. The nature of the complaints was not uniform enough to indicate fully the exact trouble. Quite a number of people were too warm, but an occasional complaint indicated a cold or cool room. By the end of the season, however, the purely mechanical troubles were eliminated and a pattern of short *heat-on* periods with long *heat-off* periods had been established.

The short *heat-on* periods were necessary because all of the radiators in any zone would at once be filled with steam whenever a zone valve opened, and if allowed to remain open for any duration of time would cause severe overheating. The long *heat-off* periods were necessary to allow time for the abnormally large inflow of heat to be dissipated. These unbalanced on and off periods caused some rooms to be cold while others were too warm since the rate of cooling varied somewhat from room to room. Therefore, in order to keep the building at a workable temperature for all occupants, the controls were set to keep the cold rooms sufficiently warm for minimum comfort requirements. This practice led to general overheating and poor economy, particularly in mild weather.

One other problem presented itself to further complicate the already bad situation; there were some occupants who were not comfortable at temperatures that were quite satisfactory to other people in the same room. While no heating engineer, unless he has prior knowledge of the exact conditions to be met, can design a system that will keep every occupant comfortable, it is the obligation of the operating engineer to make every effort in behalf of those people who are not comfortable in the normal zone of comfort. It is useless to tell a shivering or perspiring occupant that he should be comfortable because research indicates that 95 percent of all people tested are comfortable under similar conditions.

At the end of the first heating season it was apparent that the control system and steam distribution system were not in balance. Attempts were therefore made, first to adjust the orifice plate in each radiator to obtain approximately the same rate of heating throughout the zones, and second to adjust the controls so that the *heat-on* periods would be of sufficient length of time, or of sufficiently rapid recurrence, to eliminate a prolonged *heat-off* period.

Before the heating season of 1940-41 began, the control system was carefully checked and made mechanically correct. All of the steam traps were thoroughly cleaned and adjusted and all parts of the steam distribution system examined to make certain no air or water could accumulate in pockets, and that nothing would prevent the steam from readily distributing itself throughout the system. With the advent of the heating season, an attempt was made to adjust the orifice plates to obtain uniformity of heating within the zones. Little or no improvement was obtained from the attempt to adjust the orifice plates. While it was possible to affect materially the distribution of steam within the system itself, it was not possible to control the delivery of steam into each radiator to the extent of obtaining desired room temperatures.

This was due to our inability to determine how much steam would flow through an orifice at a given setting and to the difficulty of obtaining fine enough adjustments of the orifice. It was also found that some orifice plates gradually changed adjustments over a period of time. It was especially difficult to prevent the radiators nearest the zone valves from becoming hot before steam reached the last radiator off the main. However, all but the most persistent complaints about cold rooms were eliminated.

By the end of the second heating season it was apparent that in order to obtain both satisfaction and economy from the heating system, it would be necessary to have orifice plates of a fixed and very exact size in each radiator. The area of the orifice opening would have to be such that at outside design conditions just enough steam would flow through the orifice to replenish exactly the heat lost from the room. To accomplish this desired result it was necessary to know just how much steam would pass through a given orifice area at a given pressure differential, and just how much steam would be required to supply the heat needed for each room at outside design conditions.

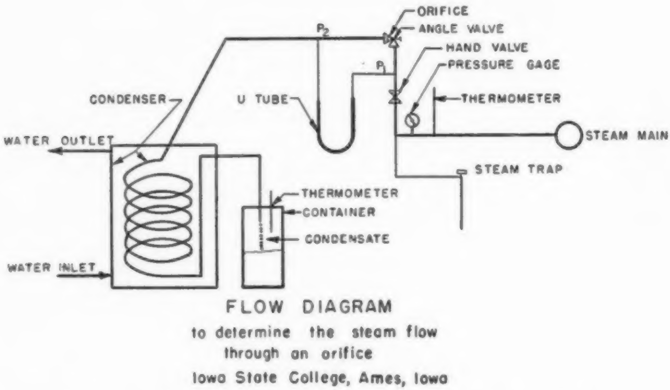

FIG. 2. TEST ARRANGEMENT FOR DETERMINING STEAM FLOW THROUGH ORIFICES

(Maximum initial pressure 5 psi gage)

The first of these requirements caused some concern. On the campus there were several heating systems in which drilled orifice plates were incorporated in the radiator valves, but for a given heat loss and steam condition there were wide discrepancies in the area of orifice used. When comparing rooms having a similar heat loss but located in different buildings, it was not unusual to find the area of opening in one orifice to be four times that of another orifice. Further, a review of published material available from books in the college library and from manufacturers' publications indicated that there were no industry-wide standards for orifice capacities. Since the quality of steam varied from very wet in mild weather to very dry in cold weather, it seemed advisable to determine the steam flow through various sizes of orifice openings at varying pressure differentials under the actual conditions of operation to be encountered.

ARRANGEMENT FOR TEST

The test arrangement used to determine the steam flow through different area openings under various pressure differentials is shown in Fig. 2. Steam

was taken from the steam main in the tunnel, passed through an orifice in the angle valve, then condensed and collected. Steam supply pressure and temperature were measured. Entrained condensate was removed through a steam trap before the steam reached the hand-operated gate valve. The orifice was inserted in a standard radiator angle valve. The condenser consisted of a coil of copper tubing in a water tank. Steam flowing downward through the coil was condensed by cooling water entering at the bottom of the tank and running to waste from an outlet at the tank top. The condensate was collected in a glass beaker.

The pressure differential, measured by the rise of a column of mercury in a U-tube, was controlled by operating the hand valve to obtain the pressure differential required. The condensate temperature was maintained relatively constant at a temperature low enough to prevent re-evaporation.

No special attention was given to the selection of drill sizes nor was any conversion of units made. The area of orifice opening was determined by simply using the drill bits that were handy at the time the orifice plates were drilled. The plates themselves were thin metal disks about 0.018 in. thick, upset on the rim to fit into the radiator valve tail piece.

Two one-half hour runs were made for each change of condition. After each test run the condensate collected in the receiver was weighed and if the results of two successive tests were not equal, additional one-half hour test runs were made until confidence could be placed in the results. The results in graphical form are reproduced in Fig. 3.

The author is fully aware of the limitations of this method of testing and the results obtained therefrom, but it must be remembered that the object was to approximate the actual conditions that would exist in the heating system, and not to make a thermodynamic analysis of the steam flow through an orifice.

In conducting the tests from which the curves in Fig. 3 were drawn, the initial steam pressure never exceeded 5 psig, this being the pressure of the supply steam. While nothing definite is known of the actual pressure on the discharge side of the orifice plate it is obvious, from an examination of Fig. 2, that the discharge pressure could not have been much greater than atmospheric.

In any case the flow conditions of the test closely approximated those encountered in the actual heating system after the orifices were installed. While no tests were made with initial pressures higher than 5 psig or lower than atmospheric final pressure, experience with other campus installations does indicate that satisfactory results can be obtained with orifice plates sized from the curves in Fig. 3 for initial pressures up to 5 psig and return pressures as low as 6 in. Hg vacuum.

STEAM REQUIREMENTS

The next step was to determine accurately the net heat loss of each space to be heated. A great deal of care was used in this step to account for every factor affecting the heat requirements. Wherever it could accurately be done, heat gains were accounted for as well as heat losses. Due consideration was given to the type of work done in each space, the activity of the people, the heat gain from steam risers, and the final temperature required in the individual rooms and hallways. Calculations of riser sizes indicated that the distribution system was not greatly oversized.

The next step was to determine the exact load each radiator was to carry. It was possible, if there were two or more radiators in a room, to admit more than the normal amount of steam to the radiator nearest a person requiring extra heat. In the case of one long room having south, west, and north

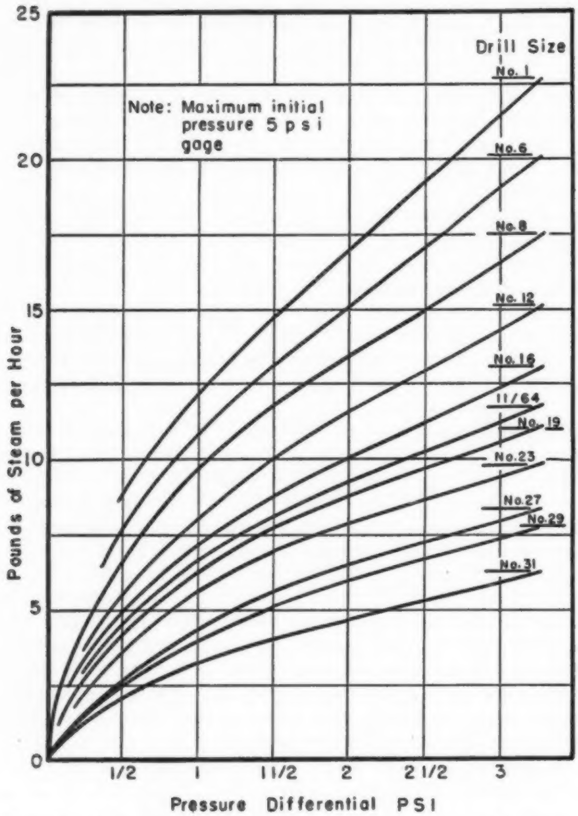

FIG. 3. CAPACITIES OF ORIFICES FOR DIFFERENT PRESSURE DIFFERENTIALS

exposures, the northwest corner radiators were given a much greater share of the load than were the west and south radiators. One radiator adjacent to a stapling machine carried very little load due to the activity of the operator.

When each radiator had been assigned its proportionate share of the Btu heat load, the required pounds of steam per hour were determined. In converting the Btu heat load to pounds of steam required, the Btu given up per pound

of steam was determined from the difference in enthalpy of the steam entering and the condensate leaving the radiator. The steam pressure at the radiator valve inlet was taken as the gage pressure on the steam main and was assumed to be dry and saturated. The condensate temperature was assumed to be 150 F, based upon the average return temperature obtained from radiators equipped

TABLE 1—TYPICAL DAILY STEAM RECORD OF SERVICE BUILDING DURING WINTER OF 1940-41

Date		Pounds of Steam Used	Degree Days	Pounds Steam per Hour	Return Temp
Jan.	7	9,814	38	258	140
	8	9,445	36	262	180
	9	10,273	45	228	140
	10	8,420	34	248	170
	11-12	13,096	59	222	160
	13	8,400	35	240	160
	14	8,804	35	249	160
	15	8,392	34	246	135
	16	8,734	34	256	170
	17-18-19	32,914	168	196	160
	20	9,840	45	219	145
	21-22	16,382	83	197	155
	23	9,534	44	218	150
	24	8,086	37	218	165
	25-26	16,480	82	201	185
	27	8,478	47	180	170
	28	8,642	47	184	180
	29	8,756	40	219	140
	30	7,869	39	201	160
	31	7,855	37	212	145
Feb.	1-2	14,905	72	207	150
	3	8,149	35	232	165
	4	9,070	35	259	155
	5	8,846	34	260	160
	6	11,508	48	241	160
	7	11,640	52	224	180

with orifices in rooms that had operated satisfactorily during the previous heating seasons.

Orifice sizes were obtained from the Chart, Fig. 3, and were based upon the pounds of steam required per hour and an assumed pressure differential of 3 psi. In actual operation this differential was reduced to 2 psi in mild weather, and raised to 4 psi in severe weather. This fluctuation of steam pressure served to maintain a building temperature cooler in mild weather than during extremely cold periods. From the viewpoint of both comfort and economy this inside temperature gradient, though only from two to four degrees, was found to be most desirable. The pressure on the return system was considered to be atmospheric.

OPERATING RESULTS

As was the case in previous years, accurate daily heating data records were maintained. Tables 1 and 2 are samples of the type of data collected for the

TABLE 2—TYPICAL DAILY STEAM RECORD OF SERVICE BUILDING DURING WINTER OF 1941-42

Date		Pounds of Steam Used	Degree Days	Pounds Steam Per Hour	Return Temp
Oct.	27		29		105
	28	5,998	23	374	110
	29	5,700	25	356	180
	30	7,690	26	495	110
Nov.	3	4,766	18	297	110
	4	4,485	25	298	105
	5	7,207	32	424	110
	6	7,652	34	477	110
	7	8,990	36	544	149
	10	6,707	31	432	110
	11	7,102	30	439	110
	12	4,250	17	265	105
	13	4,526	17	274	110
	14	4,295	18	267	135
	17	1,440	1	90	105
	18	1,452	2	91	95
	20	8,466	34	529	115
	21	5,834	29	359	115
	22	8,692	41	541	110
	23	9,328	44	582	120
	24	6,996	33	435	115
	25	5,514	26	345	120
	26	4,452	21	278	115
	27	4,028	19	251	115
	28	2,756	13	172	120
	29	3,392	16	212	110

TABLE 3—WEATHER DATA AND STEAM REQUIREMENTS OF SERVICE BUILDING FOR HEATING SEASON OF 1940-41

Degree Day Range	Average Degree Day	Average Steam Used Pounds	Pounds of Steam		
			Per Hour	Per D.D.	Per Hr/D.D.
1-15	8.8	2,391	154	272	17.5
16-30	24.4	6,190	398	258	16.3
31-40	35.6	8,695	560	244	16.0
41-50	46.2	10,015	646	217	14.0
51-60	53.6	11,600	746	216	13.9
61-up	62.5	12,566	810	204	12.9
Month	**Degree Days**	**Steam Used Pounds**	**Steam Per D.D. Pounds**	**Days Having D.D.**	**Average Month D.D.**
Sept.	212	53,095	251	21	10.1
Oct.	394	90,625	230	29	13.6
Nov.	1,123	264,050	235	30	37.4
Dec.	1,185	283,742	239	31	38.2
Jan.	1,267	273,654	216	31	40.8
Feb.	1,140	261,423	229	28	40.7
March	842	197,759	234	31	27.2
April	214	57,965	271	17	12.6
May	64	18,950	296	9	7.1

Total number of Degree Days	6,441
Total pounds of steam used	1,501,263
Average outside temperature	39.7 F
Total days requiring heat, Sept. 9 to May 24	227

winters of 1940-41 and 1941-42, respectively. These two data sheets indicate the effort used in determining the performance of the heating system and illustrate some pertinent facts concerning the operation of the heating system.

The condensate meter was read at the same time during each day listed on the data sheets. The degree days were calculated every 24 hours by averaging the hourly temperatures of the preceding day. During the 1940-41 heating season, the building was heated the same number of hours each day regardless of Sundays and holidays, but during the season of 1941-42 the heating system was operated at reduced temperatures on all Sundays and holidays. Changes were also made in the time of night shut down at the different seasons.

Perhaps the most significant daily information obtained from these data sheets is the comparison of return temperatures. The average return temperatures for the 1940-41 season were close to 160 F, while those of the season of 1941-42 averaged approximately 115 F. Since the condensate in which these temperatures were obtained included the condensate from the drip traps on the mains and risers and from the hot water heater, it is apparent that the temperature of the condensate leaving the radiators is lower. Low temperature of condensate is important because it reduces trap maintenance, reduces uncontrolled heat gains from exposed return pipes, and reduces piping strains caused by expansion and contraction. In cases where condensate pumps are used, low temperature condensate reduces possible trouble due to flashing into steam in the pump. The mechanical operating condition of the heating system itself is also reflected in the condensate temperature. A trap passing steam at even the most remote point in the system will serve to increase the condensate temperature. By careful observation of these daily condensate temperatures it is possible to detect occasional mechanical irregularities in time to prevent expensive repairs.

At the end of each heating season a summary of the heating requirements of the building was made. Table 3 for the heating season of 1940-41 is typical of these annual summaries. The data summarized on these annual sheets were used to plot the performance curves in Fig. 4. When used in conjunction with summaries of other years, data in this form yield a great deal of comparative information in a highly usable form.

Fig. 4 presents graphically the results and effect of all the work done on the heating system in the Service Building. While each year's performance curve may be compared to the calculated steam requirement, the shaded area represents the net savings over all other conditions of operation. No attempt is made here to analyze all of the information available from this set of curves. The author has never determined to his own satisfaction the reason for the flattening of the curves between approximately 35 and 53 degree days. It is interesting to note that this same curve shape exists in the results obtained from every building on the campus, for which data are available, regardless of the type of heating system.

Because the steam used for domestic hot water heating is metered along with steam used for space heating, the curves in Fig. 4 cannot be considered accurate below 25 degree days. And since there are not a sufficient number of severely cold days in the winter to make the results conclusive, the curves cannot be considered accurate above 65 degree days.

Since it is important to note the relative position of the various curves with respect to calculated requirements, something should be said about the curve

bearing the designation *calculated.* This curve is based upon the sum of the heat losses calculated for each room and is not the same as one calculated from the overall dimensions of the building. When the heat loss of an individual room is calculated, design conditions of outside and inside temperatures and

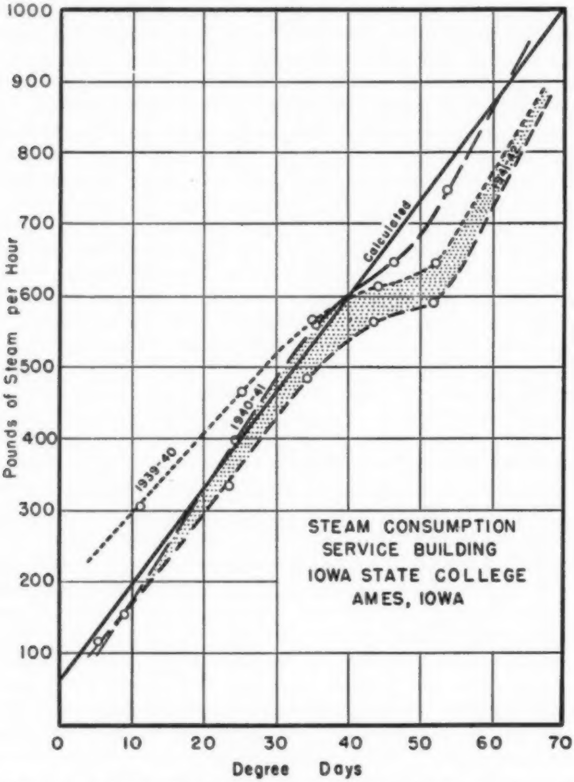

FIG. 4. COMPARISON OF STEAM REQUIREMENTS OF THE SERVICE BUILDING FOR A THREE-YEAR PERIOD

wind velocities are assumed, and the physical construction of the building is taken into account. However, when a design day occurs, it does not occur on all exposures of the building at the same time.

For example, it may be said that a building must be heated to an inside temperature of 70 F when the outside temperature is —15 F and wind velocity is 15 mph. Such a day may occur with a northwest wind, making the shaded north side of the building very close to design conditions, while at the same time the south side may be affected by the rays of the sun and be subject

to almost no wind velocity. Yet the heat required to keep the building at 70 F throughout, will be affected by every outside condition existing between these two extremes. Therefore, if the inside temperature is controlled at 70 F throughout the building, the fuel consumed will be less than that calculated for the building under the design conditions.

In the case of the Service Building, the condensate metered was collected from all of the zones prior to metering and reflected the fuel requirements of the entire building at the same instant, and therefore was less than that amount necessary according to calculations. The relative distance on the graph between the curves for actual fuel consumption and the calculated requirement will serve as an indication of the value of additional expenditures to increase fuel economy.

Additional benefits resulting from the installation of orifices in the heating system are indicated by the fact that since the installation and final adjustment of controls have been completed, there have been no troubles reported from the Service Building due to the orifices, and repairing of steam traps has been largely confined to those traps that drip the steam mains. There has been from time to time a rearrangement of rooms and usage that has required changes in room temperatures. These changes have been readily obtained by correcting the orifice plates in the radiators affected to meet the new condition.

RECOMMENDATIONS FOR ORIFICE INSTALLATIONS

The following steps should be followed when installing orifices in radiators of any two-pipe steam system:

1. Make sure that the heating system is mechanically in proper working condition, *i.e.*, that the steam and condensate lines are properly sized, pitched, and clear of obstructions. Be certain that the radiators are large enough to dissipate the amount of heat required. Make certain that the traps, especially those on the steam mains, are in good working condition, and that the system is properly vented. If the distribution system is not in good mechanical working condition, orifice plates will be of little value.

2. It is necessary that the control system be in proper mechanical condition. There must be enough control instruments, correctly located and installed, to obtain the results desired. Temperature control instruments react to ambient atmospheric conditions, not to the desires of the occupants or engineers. The importance of an adequate control system cannot be overemphasized.

3. Calculate carefully the heat loss of each heated room because the final results will depend largely upon the care used in this step. Keep a permanent record of the calculated heat losses. For the inside design temperature, use the actual temperature desired, not just an arbitrarily selected 70 F. Include heat gains as well as heat losses. The author recommends the procedure set forth in THE GUIDE published by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS for this step.

4. Assign to each radiator its proper share of the Btu load, giving consideration to the individual temperature requirements of the occupants.

5. Determine the size of orifice required to furnish the Btu output for each radiator. The curves in Fig. 3 may be used in this step. Since it will be necessary to assume a pressure differential, no harmful error will result if the gage pressure on the steam main is considered as the pressure differential if there is no positive pressure in the return system. If there is a vacuum in the return lines, the actual pressure differential across the orifice should be used. It will be necessary to use the difference in enthalpy of the entering steam and the leaving condensate to determine the pounds of steam per hour required for each radiator. However, in assuming the condensate tempera-

ture the estimated temperature of the condensate after the orifices are installed should be used. While no set temperature can be assumed, experience with buildings on the campus of Iowa State College indicates that return temperatures will seldom be more than 150 F, or less than 100 F.

6. Drill the orifice plates (several types of good plates are available) and install them in the radiator valves. The type of radiator valve used will, to some extent, determine the type of orifice plate installed. Care should be used in drilling to obtain round holes without burrs or irregularities.

7. When the heating season starts, make any adjustments necessary in the temperature controls. The types and methods of control are so many and varied that no detailed procedure can be given here, but in general the ideal condition is to have the *heat-on* periods as long or as frequent as possible, and the *heat-off* cycle as short as possible. Advice of the control manufacturer may be necessary if the conditions are unusually difficult.

8. From time to time it may be necessary to change some orifice sizes in order to make individual adjustments in room temperatures. Uneven pressure conditions in the steam mains from first to last radiator can be taken care of after the system has been put into operation.

There are some serious limitations in the use of orifice plates in radiator valves. The following three exceptions and objections are perhaps the most common:

1. If the steam used contains mineral salts, there is a tendency for deposits to form around the opening in the orifice plate. These deposits reduce the area of opening and will ultimately cause failure of the heating system. If the steam is extremely bad, the work of keeping the orifice plates clean will more than offset any advantages that may be obtained from their use.

2. Orifice plates based upon the curves in Fig. 3 should not be used in one-pipe heating systems and hot water heating systems. The orifice opening is much too small to allow liquid to flow against the flow of steam.

3. Frequently when the radiators are disconnected for painters or other workmen, the orifice plates become lost and no replacements are made when the radiators are reconnected. If even a few orifice plates are left out, a breakdown of the entire system will result.

While the correction of distribution problems is of immense assistance to temperature control, orifice plates must not be used as a substitute for a sufficient number of properly applied temperature control devices. There seems to have developed the opinion, held by many heating engineers and building owners, that the troubles encountered in a heating system are directly proportional to the number of controls installed. However, it has been the writer's experience that a lack of properly applied controls can only result in loss of both economy and comfort.

Economical heating and comfortable temperatures can result only if the flow of heat to the space to be heated is in a sufficient quantity, properly distributed, and completely controlled. The function of the orifice is to aid in the proper distribution and metering of the steam and it can fulfill no other function.

The use of orifice plates in a heating system is not a cure-all that will overcome all troubles and correct all errors of design. In fact, orifices improperly and indiscriminately used may cause more trouble than that encountered without them. The use of orifices however is based upon sound fundamental principles and when these principles are correctly applied the orifice will become a useful component of the entire system.

ACKNOWLEDGMENT

Acknowledgment is made to Gladys Bader and Don Bice for their able assistance in the preparation of this mansuscript and to Prof. John Sandfort for his generous advice and counsel.

DISCUSSION

E. F. HYDE, Detroit (WRITTEN): This is a fine contribution to our knowledge of orifices. It is especially valuable to know of the satisfactory results which have come about from their use, both from the standpoint of comfort and of economy of operation.

The work of adjusting an orifice system is a laboriously long process. A method of adjusting, such as the author suggests, should be used if the needed results are to be obtained. The method of testing the orifices to determine the steam flow through them is unique and should serve for the application of orifices to any system employing steam above atmospheric pressure.

The comparison of steam requirements for the building tested as given in Fig. 4 over a three-year period is a very interesting way of presenting the results. The flattening of these curves between 35 and 53 F is surprising and it is to be hoped that the author will be able to account for this phenomenon later. From a practical standpoint it would also be valuable to know the overall annual steam consumption of the building so that results could be evaluated on a monetary basis.

From a consulting engineer's viewpoint in the choice of a type of heating system, there is a question as to whether an orifice system should be employed. The sub-atmospheric system would no doubt minimize the necessity for orifices and the results would be obtained almost immediately. If the cost of adjusting an orifice system were taken into consideration, it would seem that the sub-atmospheric system could be installed for less money than an orifice, and the economy of operation might be expected to be even better.

In connection with an orifice system it is noted that the author recommends that steam and condensate lines be properly sized, pitched, and clear of obstructions. These precautions are no different than for a sub-atmospheric system and should govern the installation of any type of system. The question of whether orifices or some other means of equalizing flow should be used, however, should not in any way detract from the value of this interesting paper on the use of orifices.

M. W. SHEARS, Toronto, Ont., Canada: In our experience in orifice application we have found it necessary to go farther than the author in determining the size of orifices. We have found it necessary to allow for the pressure drop in the steam lines by providing larger orifices for radiators farthest from the source of steam supply. We determine orifice size from tables that we have established through experience.

W. A. GRANT, Syracuse, N. Y.: We are all curious about this hump in the curve. If you will do a little mental arithmetic, you will see that it occurs at the time that most people are wont to open the windows rather than control the radiators. I am wondering whether that partially explains it.

G. D. WINANS, Detroit: There are on the market several proprietary heating systems using orifices for obtaining proper distribution of steam. Some basic information on this subject was presented in 1931*. Orifices for the regulation and distribution of steam and hot water have been in use for many years. In 1885 an English patent describes the use of orifices to obtain proper distribution of steam in heating systems.

* Flow of Steam Through Orifices into Radiators, by S. S. Sanford and C. B. Sprenger (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 371).

J. W. JAMES, Chicago: My comments relate to the overall improvement in comfort results which may have been achieved as a result of this orifice installation.

Departing briefly from a consideration of orifice regulation for commercial and large building installations, I would like to comment on a related subject common to small automatically-fired steam systems for residences. In this type of system the problem of obtaining adequate and proper steam distribution has received considerable attention. Basically the problem has been one of obtaining uniform and proper steam distribution to the last radiator on the piping system as soon as possible after the thermostat calls for heat.

It is appreciated that the idea of orificing a residential steam system is not new, but I wonder whether previous consideration of this idea has emphasized the possibility of obtaining better steam distribution with orifices. True, variable radiator air vents have been used to accomplish this objective, but in some instances complete knowledge concerning their application to obtain adequate steam distribution has not been utilized. Hence, it seems possible that additional research might be projected to fully investigate the possibilities of the use of orifices to obtain better comfort results in small steam systems.

Another comfort problem associated with these small steam systems is the interval which elapses between the time the thermostat calls for heat and the time steam enters the first radiator. With the advent of automatic dishwashing equipment, temperatures of 180 F are now required, which means that the boiler water temperature regulators will have to be increased to 200 F in order to obtain the necessary hot water. With boiler water temperatures uniformily maintained at this temperature it would require only a short interval for steam to be generated, which should be a further step toward obtaining improvement comfort.

As most of my comments have dealt with comfort, I am anxious to learn from the author whether, as a result of the revised orificing system described in his paper, the desired comfort results were achieved? In other words, in those rooms requiring 80 F, were these temperatures obtained, along with adjacent rooms requiring 70 F, as a result of the changes made with the orifices? If the results were favorable on this large building system, perhaps comparable results could be obtained on the small residential system.

The author mentioned that one of the problems associated with the use of orifices on a steam installation was the build-up of precipitates at the small orifice, which required some maintenance. If this became a serious problem it would seem that modern boiler water treatment with chemicals could overcome this problem adequately.

M. F. BLANKIN, Philadelphia: Webster Tallmadge, a member of the Society who died recently, did a great deal of work on orifice application. I do not know whether any of his work was ever put in written form or whether it was ever tabulated in any way, but he used possibly 100 to 150 different orifices on any one system, developing what he called a metering type, as he claimed he could make them accurate to the nearest square foot.

He also had a method of control by which he started with 48 oz of pressure and carried it down in a series of steps of about 10 percent reduction each. He claimed very accurate results for such a system. I think his original work on that was done as far back as 25 years ago.

As far as the small system that John James mentions, we have done a great deal of work in applying such a system to jobs from 10 radiators on up. For distributional purposes only, it has been very satisfactory.

C. F. MALLY, Detroit: There are several things that could be explained further in this building. The thing that occurs to me is: How did the final sizing of the radiators, after orificing, compare with the initial design of the radiators? Were they

much too big or just about as calculated? Also, what was the basis of the original zoning of the system; were the outside exposures taken care of somewhat reasonably and the inside rooms with the hope that they might be shut off if it was too much?

AUTHOR'S CLOSURE: In regard to Mr. Shears question, we have found it necessary, especially on larger installations, to take into account the pressure drop in the piping system when determining the orifice sizes. I have found on occasion, however, that this pressure drop is not too important when a modulating type of control is used. When this type of control is used, it becomes necessary to close the control valve when the pressure in the distribution main reaches such a low value that steam cannot reach the far radiators.

Experience indicates that the limiting pressure differential beyond which distribution fails is much lower than calculations indicate. We feel that this is due to the condensing action of the steam in the radiators and piping.

Concerning Mr. Grant's question about the opening of windows, there were occasions when some occupants opened windows and the results were, of course, the lowering of room temperature. Whether or not it would be possible to design a system in which no one would open windows without actually installing windows impossible to open, I do not know. There are no buildings on the campus of Iowa State College in which it is possible to satisfy all of the people to the extent that someone will not open windows. The best we can hope for is that when windows are opened, it will cause discomfort to the fewest people and poor economy to the smallest degree.

In the case of the orifice installation, an open window did not affect the economy and whether or not it caused discomfort to others, I do not know. If there was discomfort, it was not to the extent that any complaints were registered.

I believe that Mr. Winans might be interested in knowing just how much background we had before any attempt was made to determine the steam flow through the various orifice plates. We have on the campus of the Iowa State College a fine technical library. Before we started to determine the steam flow through an orifice, the library staff gave me a file of information on the subject. Part of that information included the work of S. S. Sanford and C. B. Sprenger. We found information on steam flow through an orifice as far back as 1901. It was as a result of what we thought was the inconsistency and inadequacy of this information that our determinations were undertaken. We do feel, therefore, that we had adequate reason for undertaking our work, and we hope that the results of this work will be of value to others.

I was glad to learn of Mr. Blankin's success with orifice plates when used in small heating installations. Unfortunately, my experience has been limited to buildings not greater than eight stories and predominantly of three and four stories, and so I have never been able to determine the upper limits on building size for an orifice installation.

I would like to point out to Mr. Hyde that the work of making an orifice installation is not necessarily a laborious proposition. I have recently finished an orifice installation in an eight-story hotel building in which all of the engineering work was done in two days. In the case of the Service Building at the Iowa State College, the long time involved was spent in gathering data on the steam flow through various orifice openings.

If the building is large and there are many rooms, it is well, when calculating the heat loss, to set up a series of curves by which the heat loss through walls and glass and various other items of heating loss can be taken directly from the curves. In this manner, the problem of calculating heat loss is greatly simplified.

It is difficult to give an actual monetary amount when comparing orifice installation with other types of heating systems, since the amount of money saved by an orifice installation is dependent largely upon how bad the system was before orificing. On those orifice installations with which I have had something to do, it has so happened that the cost of engineering was paid for the first year of operation.

I have never made a comparison of an orificed system with a sub-atmospheric system since I have never felt that the two systems were exactly comparable. My experience leads me to believe that orifice plates are of value when installed in a sub-atmospheric system since the purpose of the orifice plate is to assist in the control of the distribution system. In other words, the fact that a system operates at sub-atmospheric pressures does not itself mean that proper distribution will occur. Whether or not an orifice system would be more or less expensive than a sub-atmospheric system, it seems to me would depend upon the factors of size, location, method of supplying steam, etc., that surround the individual job.

With regard to Mr. James' inquiry whether the desired temperatures which were different for various rooms had been automatically maintained after the orifices had been installed in the system, the answer is that, taken over the entire heating season, they were not maintained at the exact calculated value. The reason for this appears to me to be that design conditions are never attained on the entire building at any one given time.

If we would assume a room on the north side of the building and design for a given set of conditions, that room would and did have the desired temperature when those conditions existed. When, for instance, the wind was from the south and did not greatly affect our assumed room on the north side of the building, the temperature in that room increased somewhat. The heat migration through the building, due probably to atmospheric pressure differentials throughout the building, also tended to affect the temperature of individual rooms throughout the year. Actually, a given room would vary as much as three degrees from design conditions, but since the type of occupancy did not require absolutely uniform control, no harmful results occurred.

Mr. James' comment about the use of orifices in small systems is in complete agreement with my own experience. I do not know exactly how small a *small* system is in Mr. James' mind, but I believe orifice plates installed in a heating system will assist in the distribution of any size heating system since the function of the orifice plate is to deliver the proper amount of steam to each individual radiator in the system. We have had very excellent results in controlling heating systems in six family apartments from a single thermostat. We have also found excellent results in eight-story buildings of 300 rooms.

Mr. Mally's question about obtaining design conditions in actual performance, can best be answered by saying that when outside design conditions occurred, we obtained assumed inside design conditions; but since outside design conditions or their proportionate amounts are rarely obtained for any given room during the heating season, and never occur for the entire building at one time, it must be stated that the inside design conditions deviated from the assumed design conditions. However, this deviation was in the nature of two or three degrees and caused no appreciable amount of discomfort.

One of the conditions that caused the widest variance seemed to be heat migration from the windward to the leeward side of the buildings. This migration, of course, is a problem that presents itself in all types of heating systems. But, again, it did not contribute enough extra heat to cause general overheating on the leeward side of the building.

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No. 1397

PLANT RESEARCH IN THE PHYTOTRON

By A. J. Hess*, Los Angeles, Calif.

A PROJECT now being carried on at the California Institute of Technology is concerned with the study of the most common and the simplest solar energy gathering machines—growing plants. It is well known that plants, by the process of photosynthesis, change the sun's radiation into chemical energy and lock it into the plant in the form of starches and sugars made from water and the free carbon dioxide in the atmosphere. The process breaks down the CO_2, discarding the oxygen back to the atmosphere and using the carbon and water to make up the starches and sugars. Thus many ordinary field plants store up huge amounts of solar energy in a few months, which can be released in two ways of interest to the engineer: (1) direct combustion of the plants to produce heat; (2) conversion of plant sugars and starches into alcohol, a good fuel which can be stored, transported, and utilized in existing facilities.

Another important subject under study at the laboratory is food production. With an expanding world population, the problem of supplying food for all has become increasingly important. This project, which has been going on at the California Institute of Technology for the past 11 years, involves the *phytotron,* so named because it performs the same function in plant research that the cyclotron does in nuclear research. The problem of increasing food production can be solved by: (1) increasing the efficiency of growing plants; and (2) by increasing the efficiency of agricultural processes. Almost all agricultural progress has been in cultivation equipment and cultivation methods with some progress in plant breeding, but knowledge of the actual growth of plants, their photosynthetic efficiency, and their dependence on climate and weather is little changed from prehistoric times.

Contributions of the Phytotron

Work at the phytotron has shown that the function of the soil is minor compared with that of the atmosphere and sun. A few years ago such a statement would have been surprising, but it is now known that the major function of the soil is to provide support for the plant structure, meter moisture to the roots

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* Member-Owner of Hess-Greiner & Polland. Member of A.S.H.V.E.

Presented at the Semi-Annual Meeting of The American Society of Heating and Ventilating Engineers, Muskoka, Ontario, Canada, June 1950.

FIG. 1. ARCHITECT'S DRAWING OF EXTERNAL VIEW EARHART PLANT RESEARCH LABORATORY

and supply the small amount of minerals required by the plant. All of these soil functions have been done in the laboratory with a synthetic sterile medium in which the roots can grow. This involves the now familiar science of Hydroponics: wherein plants are grown with roots immersed in nutrient solutions without any soil or solid media whatsoever for root growth.

The laboratory at the California Institute of Technology is built to study all the environmental conditions of plant growth both for roots and superstructure. Studies to date establish the fact that the average field plant is about ½ percent efficient photosynthetically. Work now in progress at the laboratory indicates that this can be increased to 4½ percent, a low figure, but still nine times the efficiency now obtained on the farm. Little need be said concerning the tremendous political and economic significance of these figures, which would mean that with the same growing facilities including land, machinery, labor and water, the nation could increase its output of food nine times.

Medicine: Another potential accomplishment of this research project indicated by studies to date is its contribution to medicine. Many diseases and ailments of the body result from a deficiency of some chemicals used directly in the body structure, or as catalysts to control natural functions. In their natural state which is usually inorganic, many of these chemicals cannot be assimilated by the body, but if these chemicals are fed to plants first and thus made to take on an organic character they can be assimilated easily by the body. The study of the methods of feeding plants the desired chemicals was discontinued about 1941, because of a shortage of technical personnel and funds.

A drug to control high blood pressure, a major mortality cause, has recently been found in a wild plant. The new drug can be extracted from the plant, but the plants are so few in number and are located in such inaccessible areas that practical production from natural sources is impossible. At the phytotron laboratory, a study of the environmental requirements of this plant is now going on. It is hoped that with the data obtained this new drug can be produced commercially in the United States.

Seed Control: An additional important result reached by the laboratory has been to establish a complete control of seeds by the elimination of all sources of flower contamination due to errant pollens, seeds, yeast spores and mold spores either air borne or insect borne. Thus scientists will be able to crossbreed plants with accurate ancestral records together with complete control of all genealogical factors. It is probable that within the next 20 years all seeds will be grown under such controlled conditions, requiring large greenhouses equipped with air conditioning.

Smog Research: Air pollution or *Smog* research is also an important part of the work at the phytotron. Human nature is one of the major problems in investigating air pollution because of individual differences regarding reports of exposure to polluted air. Plants, on the other hand, are secure in one position and are unable to talk and therefore make good subjects for use in *Smog* investigations. This was accidentally discovered in the phytotron when it was found that purging of refrigerant gas which was leaking into a condenser caused plant damage simultaneously with the purging of the condenser. This experience led the scientists in charge of the laboratory to conclude that many gases in the atmosphere existing in minute quantities could be investigated with respect to plant damage and also with some respect to effects on humans and animals.

HISTORY OF THE PHYTOTRON

It would be difficult to give a complete history of the phytotron without an acknowledgment to those scientists both great and near great who contributed over the centuries to the small fund of knowledge on the subject of growing plants. The acknowledgment of modern times is due Dr. H. O. Eversole of La Canada, Calif., who started the modern and accelerated research into this subject. Dr. Eversole is an orchid grower who built an air conditioned greenhouse to reproduce the native growing conditions (including both day and night) of orchids by which he obtained phenomenal growing results. Others became interested in the project, the first being the U. S. Department of Agriculture. In 1931, the Department under the direction of Dr. L. C. Marshall, a physicist, constructed an air conditioned greenhouse at La Jolla, Calif., and obtained valuable data on Papaya plants.

Air conditioning engineers showed keen interest in the La Jolla project but, strangely enough, not because of the air conditioning for plants, but because it was one of the few successful heat pump installations then in operation. From the results of these two installations, Dr. Eversole in 1938 raised funds and persuaded the California Institute of Technology to construct the first unit of the phytotron. The project was placed under the direction of Dr. F. W. Went of the Biology Department at the Institute. With the small pilot laboratory, he obtained such important results that the Earhart Foundation of Ann Arbor, Mich., supplied funds for the construction of the main laboratory now in operation.

The project is now beginning to awaken world scientific interest and similar laboratories are being built or contemplated in South Africa, Canada, Belgium, and India with arrangements for interchange of scientific data and personnel.

PURPOSE OF THE PLANT LABORATORY

The phytotron (Fig. 1) is a plant research laboratory designed to control all the environmental conditions of growing plants for both superstructures and root systems including: control of temperature, humidity, and sterility of the air; the food supply, the chemical and sterile quality of all water supplies; all light sources both as to intensity and spectral qualities. Every precaution is taken to maintain all of the environments sterile after originally sterilizing them.

MECHANICAL ARRANGEMENT OF THE PHYTOTRON

A survey of the requirements of a phytotron indicates that a mechanical engineer would find a real professional interest in the design of such a project, since it is essentially a series of air conditioning systems with a building around them. When this is complicated by the usual request from the owner to design twice as much laboratory as available funds permit, the engineer is faced with a problem.

The phytotron, though only a small building approximately 80 ft × 125 ft with a ground floor (Fig. 2) and basement (Fig. 3), contains the following:

a. Six air conditioned greenhouses for growing plants under natural sunlight, with provision to maintain day temperatures from 58 F to 90 F and night temperatures from

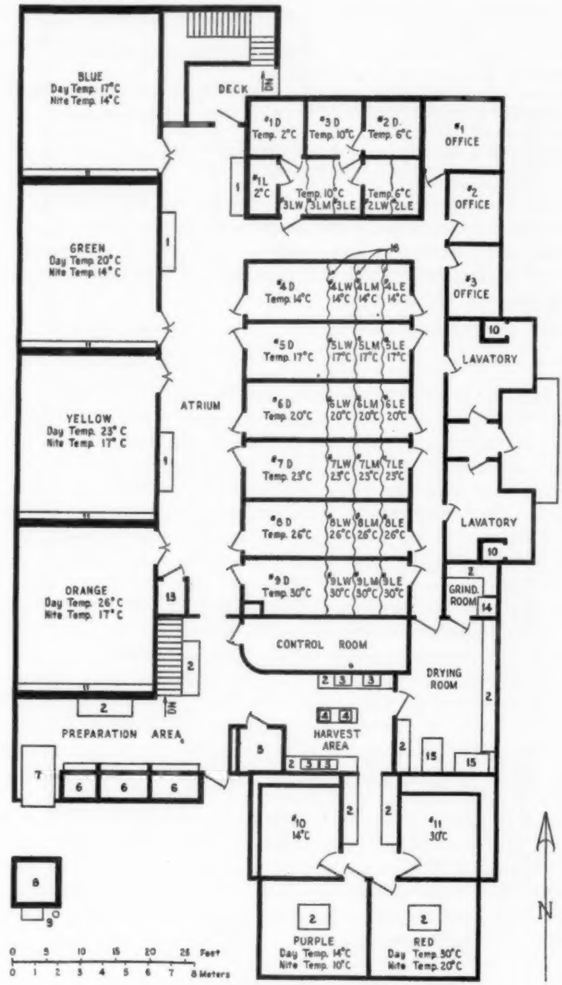

FIG. 2. PLAN OF LABORATORY GROUND FLOOR

50 to 68 F. All these temperatures may be combined with any desired relative humidity from 15 to 90 percent.

b. Thirteen air conditioned laboratories provided with artificial light and equipped to maintain conditions from 35 F to 90 F with any relative humidity from 15 to 95 percent and with light intensities up to 1500 C.P.

c. Eleven dark rooms all equipped to maintain 35 F to 90 F with any relative humidity from 15 to 95 percent.

d. Ten laboratories of various sorts all air conditioned as required:
Five laboratories that are light conditioned
One laboratory for maintaining zero dewpoint with temperatures of 40 F and above
One for radioactive experiments
One for all types of gas experiments
One equipped with a wind tunnel for wind experiments
One with provisions to produce fog or rain.

In addition, the phytotron is equipped with harvesting, grinding, and drying rooms; a chemical laboratory for the analysis of the harvested plants; a photographic studio and laboratory; and receiving and potting room with sterilizing equipment for all entering plants and materials. These systems require 21 air conditioning units of various types.

EXPERIMENTAL CONDITIONS

In order to provide flexible operation for the relatively large group of plant researchers, the phytotron is designed so that each individual laboratory is set up at a predetermined condition and the plants are moved from laboratory to laboratory in accordance with the schedule set up by each researcher. This enables each worker to set up his own schedule of temperatures, humidities and light exposures, including time of exposure to the various conditions.

Thus each laboratory can be used simultaneously for many experiments and by several workers, the only limitations being those of available space and in some rare cases the possibility of undesired crossbreeding. This system of operation does pose a bookkeeping problem since records of conditions and time of exposure to conditions must be kept for each plant. Thus, one of the important departments in the phytotron is the bookkeeping department.

The advantages of this method of use are illustrated by a successful experiment made to determine growing times of temperate climate plants. Exposure of the plants to 4 hours of equivalent daylight and 8 hours of darkness revealed that the plants would grow to maturity twice as fast as normal plants—each 24 hour day being converted into two growing days. This was done by moving the plants from laboratory to laboratory on a schedule.

The method of operation outlined requires that plants be grown on movable carts instead of conventional greenhouse benches. These carts are shown in Fig. 4. Each cart will hold six plants in gallon cans or four plants in two gallon crocks. The carts are moved from room to room by means of a small electric locomotive which pulls the carts around the building like a string of cars behind a railroad locomotive.

COOLING REQUIREMENTS

The cooling required by the systems is generated by three reciprocating compressor units using Freon 12 refrigerant and equipped with water cooled condensers. One small greenhouse with a dark room and artificial light room is equipped with one of these condensing units, a 5 hp machine supplying a direct expansion coil to provide independent operation of this house. Those rooms, which can be maintained below 50 F if desired, are equipped with a 7½ hp condensing unit supplying refrigerant to three air conditioning units using

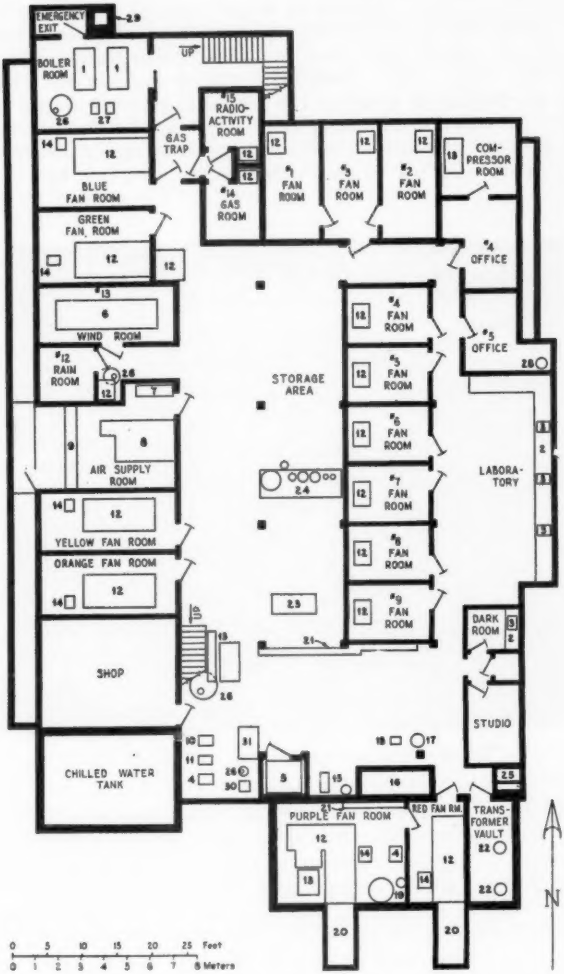

FIG. 3. PLAN OF LABORATORY BASEMENT

direct expansion. This was done because in the lowest temperature rooms the refrigerant can at times be below the freezing point of water, which excludes the use of the main chilled water system.

The other laboratories are supplied with chilled water cooled by means of a 60 hp condensing unit supplying refrigerant to water chillers. The chilled

water is stored in a 30,000 gal tank for a total capacity of 110 tons during any four hour period. The great diversity of use of these laboratories indicated the requirement for a very flexible system, and chilled water was selected as the best answer to this requirement. Centrifugal pumps circulate the chilled water throughout the building at a rate of 150 gal per min (gpm) with a peak of 250 gpm. There the water is utilized by remote type air conditioners for each air conditioning system as required, and is also piped to supply and return outlets in each laboratory for use in root temperature and water bath experiments as desired.

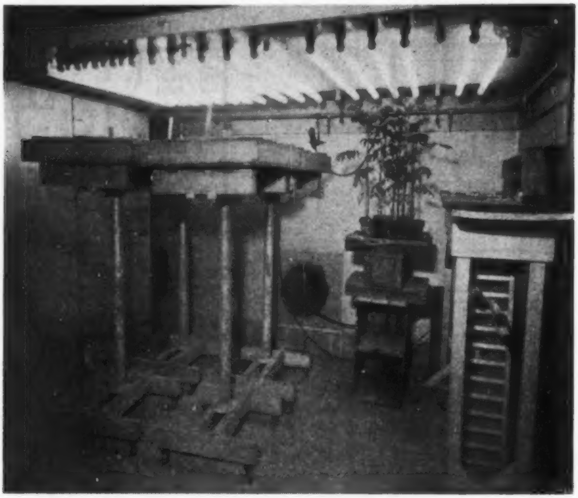

FIG. 4. SPECIAL CARTS FOR MOVING PLANTS

Condenser water is supplied from a roof spray system, using water which has been allowed to flow over the roof of the greenhouses to accomplish sun radiation control in addition to supplying water for condensing. The condenser water is recirculated with centrifugal pumps and is passed through a sand and gravel filter after treatment by bromine to prevent algae growth. The rate of condenser water circulation is 195 gpm. At this rate an approach of 5 deg to the wet bulb temperature was obtained, though in addition to the condensing load, there is a load of approximately 90 Btu per square foot of glass roof due to absorbed direct and indirect radiation.

HEATING REQUIREMENTS

Heating is provided by two cast iron, gas fired boilers used as direct fired water heaters. Centrifugal circulating pumps circulate the hot water throughout the building where it is used as required by the remote type air conditioning units and it is also piped to each laboratory for root or water bath temperature

experiments. The heating plant has no direct entrances or accessibility from the laboratory spaces, thus eliminating any possibility of contamination of laboratory air by products of combustion.

Steam for sterilizing is supplied by a small boiler. Tools, pots, and other hard goods are sterilized in a hospital type sterilizer with steam jacket or direct injection where possible. Plants are sterilized with triethylene glycol vaporized by a steam coil while sand, gravel, etc., are sterilized by direct steam injection through perforated pipes. It is interesting to note that even the cigarettes of the laboratory workers are sterilized.

HUMIDITY CONTROL

Humidity is controlled in three different ways. In the greenhouses, dehumidifiers with preheater coils control the dewpoint of the air and a reheater is used

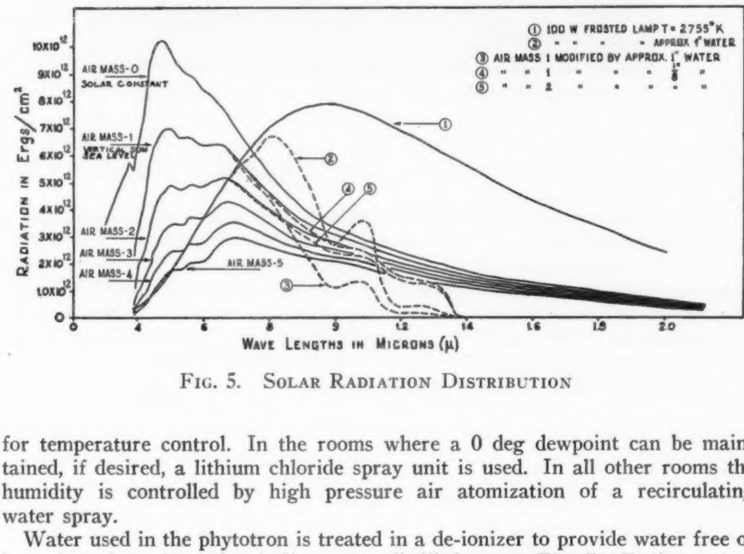

FIG. 5. SOLAR RADIATION DISTRIBUTION

for temperature control. In the rooms where a 0 deg dewpoint can be maintained, if desired, a lithium chloride spray unit is used. In all other rooms the humidity is controlled by high pressure air atomization of a recirculating water spray.

Water used in the phytotron is treated in a de-ionizer to provide water free of bacteria and nearly as chemically pure as distilled water. The distribution system is of hard rubber and all storage tanks are of glass or are rubber lined to assure uncontaminated water at the point of use.

STERILIZING LABORATORY AIR

The sterility of the air is maintained (insofar as pollens, yeast molds and insects are concerned) by taking all outside air in at one source and electrostatically filtering at the outside inlet before introduction into the various systems. A circulating blower with static pressure control maintains the flow of treated air to all air conditioning units where outside air is required. The laboratory is under a positive air pressure at all times. There are only three

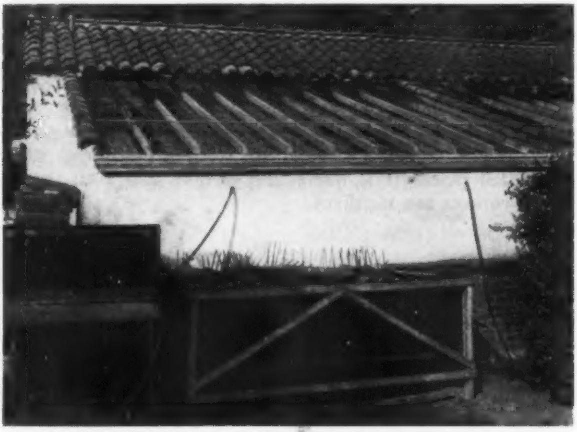

FIG. 6. ROOF SPRAY ARRANGEMENT FOR PART OF LABORATORY

weather stripped doors to the outside and all windows are sealed to prevent the entrance of contaminated air.

Nutrient (plant food) solution is mixed in a mixing and storage tank and is circulated throughout the building by means of a centrifugal pump to outlets in each room, where solution strength can be adjusted by dilution with de-ionized water and temperature can be adjusted with a chilled or warm water bath.

Compressed air of various pressures is supplied to each room from a central compressing unit. This air is for such general use as agitation and cleaning, and is taken from the discharge side of the electrostatic air filter.

In the course of working on the phytotron, many technical features of greenhouse air conditioning have been verified and can now be made available for use of design engineers. Some of these technical features are of use in general

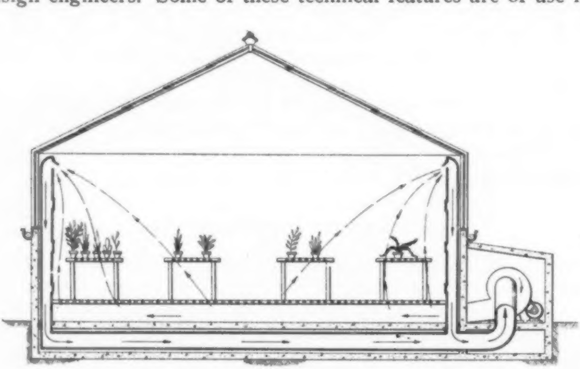

FIG. 7. AIR DISTRIBUTION SYSTEM FOR LABORATORY

greenhouse design and some are at present only of use in the design of research greenhouses.

CONTROL OF RADIATION

The most important single factor in greenhouse design is *control of radiation*. It is important because the process of photosynthesis (the process by which plant life converts carbon dioxide and water vapor into carbohydrates) is dependent on sun light, the radiation from the sun. Studies of sun light indicate that a total of 1800 Btu per sq ft per 14 hour day or approximately 200 Btu per hour per sq ft at the mid-day peak will reach the average roof on an average mid-summer day in the temperate zone. Roughly 96 percent of this radiation will pass through a glass roof unless intercepted in some manner. It has been the custom to intercept much of this load by painting or shading the glass, but shading or painting unfortunately also reduces the rays from that part of the spectrum which are good for photosynthesis. Consequently, though temperatures can be held to levels which prevent plant damage, the growth of the plants will likewise be held to a low rate.

Further study has revealed that much of the sun's light is not used by plants and as a matter of fact even causes some plant damage. These non-usable rays are in the infrared portion of the spectrum. Experience has determined that a film of water 1/16 in. thick will remove 19 percent of the sun's rays and most of this percentage is infrared.

The solar distribution curves based on publications of the Smithsonian Institute illustrate this point with respect to the total radiation received from the sun by the earth's atmosphere (see Fig. 5). Note the effect of thickness of water in absorbing the radiation and also that practically no radiation of wave length above 1.3 millimicrons (1,300 angstroms) is transmitted. This is all in the infrared band. Reflection from the two water surfaces and two glass surfaces, added to the amount of heat intercepted by the cooled under surface of the glass at normal wet bulb temperatures of 76 F or less, is about 25 percent. Thus, a total of 25 percent plus 19 percent or about 44 percent of radiation can be removed by the film of water on the glass surface.

These facts, verified in the laboratory, show that a water film will allow the plants to grow at close to the normal rate of growth while holding temperatures at an even lower level than with painting or shades (and it is well known that painting or shading of the glass will practically stop growth). In the phytotron there is a saving of about 275,000 Btu per hour on the cooling load by use of the water film. Since most glass roofs are sloping, the water can be sprayed on near the ridge and allowed to flow down the glass (Fig. 6). At the phytotron, double use of this water is made—it is the condenser water and also a radiation shield. A circulation of about 1½ gpm minimum per foot of glass width seems to give good coverage.

So great is the radiation load on greenhouses that for cooling of plants of tropical, subtropical and temperate climates, the only cooling load that need be normally considered in the design calculations is the radiation load. All interior surfaces in greenhouses should be painted with aluminum paint to give as equal a distribution of radiation as possible. The roof spray water must also be properly treated for control of scale and algae.

Another important environmental condition is temperature. Whether the plant flowers properly and bears good fruit is almost entirely dependent on a proper range of temperatures at some time during the life cycles for annuals or reproduction cycle for perennial plants. However, it is impossible to set up design temperatures for plants since each plant will require a different condition.

The engineer should check the native habitat of the plant for the conditions best suited to the plant. His design should include provisions to reproduce proper day and night conditions (the night conditions being the most important since actual growth occurs at night and the rate and quality of growth and fruit production are determined by night conditions).

The designer should avoid using design conditions set up by nurserymen who have not determined the plant's native environment. Nurserymen base their data on information gained as a result of experience but without observing the thermometer, and though most nurserymen may feel that proper conditions for plant growth exist, they fail to relate their experience to the actual scientifically determined conditions.

Except for certain plants, such as the succulent family, which grow under extreme humidity conditions, humidity is not so important to most plants as was originally thought. Desired humidity design conditions should be determined as carefully as the desired temperature conditions. In the study of humidity effects, one family of plants was found sensitive to relative humidities exceeding 75 percent. This plant had a leaf with a practically vaporproof top; the underside of the leaf absorbed moisture from surrounding air when the relative humidity exceeded 75 percent but closed its moisture absorbing units in the leaves to prevent loss of moisture due to reverse action when the relative humidity fell below 75 percent. Planted on the desert in quantity, this plant could absorb moisture during nights when the desert temperature usually approaches the dewpoint and thus ultimately makes the desert bloom by feeding the plant roots and indirectly the soil with moisture removed from the air during times of high relative humidity.

CONTROL OF AIR FLOW

Air distribution is a very important design factor. In most greenhouses the combination of a high radiation load, a relatively low temperature and moderately high humidity combine to require high air volumes, usually in the range of a one minute air change. At the phytotron the change occurs in 20 to 40 seconds. Further, since overhead distribution casts shadows and because plants do not like drafts, low air inlets and high air outlets are desirable. This upward moving air containing water vapor will also remove some long wave length radiation before it reaches the plants. At the phytotron, workers experimented with, and are now using, a unique method of air flow through the floor using a plenum space under the floor with air inlet slots through the floor (Fig. 7). Slots should be ¼ in. or less in width, and velocities up to 2000 fpm can be used with good results. In operation now are several greenhouses using slots in the floor with good results.

Many other factors are still under study. In the meantime, the air conditioning profession should keep an eye on this type of research not only for its benefits to mankind, but also for its interest to engineers because they will be

called in to design huge seed production greenhouses and many large greenhouses in which luxury crops will be produced regardless of season.

DISCUSSION

P. J. MARSCHALL, Chicago: I do not recall reading the words *sterile air* and do not understand how you obtain sterile air with the type of filters used. When our bacteriologists want sterile air, we are required to filter the air through carbon filters as we have not found any other type of filter which will remove mold spores.

Also, I would be interested in knowing how closely you control both temperature and relative humidity.

E. R. QUEER, State College, Pa.: Perhaps the lack of comment is because the author has covered the subject so well. One thing I would like to comment on is the fact that insects are very quickly gotten rid of in the supply fans of a ventilating system. For years the Navy had a custom that all supply fans had to be screened. You can readily appreciate the difficulties we had with those screens continually being clogged with dirt. After some experiments were run, it was proved conclusively that the fan is an effective fly swatter.

F. C. HOOPER, Toronto, Ont., Canada: It is interesting to compare the attainable solar energy collection efficiency of the photosynthetic process with that of direct solar heat collectors. The author points to a possible 4½ percent maximum efficiency from living plants. Presently practical solar heat traps of the type used for solar house heating, and the hot water and steam producing mirror type absorbers will give efficiencies in excess of 50 percent.

This is not to say that the solar heat trap is directly competitive with the field plant, for the latter produces a highly available store of chemical energy, but it does emphasize the high order of ground area utilization possible with heat traps used for low temperature heating service.

S. W. BOYD, Atlanta: In what manner is the increased growth efficiency of the plant manifested? Is the size or the caloric content of the plant increased? What effect does a change in the length of the day-night cycle have upon the growth efficiency?

D. W. LOCKLIN, Cleveland: How is the efficiency of a growing plant defined and how is it determined?

R. A. MILLER, Pittsburgh: I was particularly interested in the question of radiation and radiation control and, more specifically, in the use of the very thin film of water over the glass surface. The radiation which is permitted to pass through most of the glass which we use, except very special types, is very limited in the range of transmission, materially lower than the solar energy that would be available at the top of Mount Palomar but well within the range available at ordinary sea level or living levels in the world.

Water primarily absorbs infra-red, I believe; it takes in very little of the ultraviolet; it permits it to pass through quite freely. Ordinary glass will transmit from 0.292 millimicrons wave length to 2.92 millimicrons wave length or, in terms of angstroms, 2920 angstrom units is the maximum, 292 angstrom units is the minimum. And the range of transmission through various types of glass can be made to vary from zero at 400 angstroms to a maximum at 550 or it can be exactly reversed and have a minimum at 550 and a maximum at 400. Thus the control of radiation to the plants can be almost absolute, according to the desires of the person building the greenhouse and the types of glass that he is willing to use to glaze that roof.

He can get glass which will not be subject to breakage under hailstorm or he can get glass that will fail under the first impact of a tiny stone.

The radiation control can be made to conform exactly to that radiation curve which you wish to obtain.

I do not like windowless rooms; however, the growing of vegetation in the dark might be a most interesting proposition. After all, we have a great many subterranean workings where men have to live and also a great many subterranean chambers where many plants grow. I presume that knowing how plants will grow in the dark and how they will ameliorate the conditions under which human beings will live are important.

CARL H. FLINK, New York: Mr. Miller has pointed out the possibility of controlling radiation to plants by use of the proper kind of glass. In cases where a variation in radiation might be desirable, perhaps the water spray might be advantageous because it could be varied.

AUTHOR'S CLOSURE: First, with regard to Mr. Marschall's comments on obtaining sterility of air without carbon filters, he is exactly right. We found that it does require carbon filters and hence we have two of the laboratories which we have equipped with a standard, commercial, activated charcoal filter. We found that it took three in series to get the sterility that we wanted.

As to temperature and humidity, each laboratory is set up for specific conditions. We try to maintain the humidity within 3 percent, the temperature within a degree on either side of the control point. The plants are moved from laboratory to laboratory to get the conditions of plant growth that the particular research worker wants. To accomplish this, plants are not grown on benches but on little carts that are hauled around on a predetermined schedule set up by each particular research worker.

With regard to the plants as heat traps, we acknowledge that probably a more efficient instantaneous method of trapping sun radiation can be used, but the advantage of the plant is that it is cheap and works over a period of six months or as long as it takes the plant to grow, consequently, we are accumulating for ultimate harvest energy from all periods in which energy is available in any amount and there are no limitations due to dark days or lack of sunshine for protracted periods.

As to growing in the dark, we do grow the plants in dark rooms using artificial light, the idea being to show that we can grow in any season of the year. Heretofore, people thought that greenhouses could grow plants without regard to the weather, but that is not true. Greenhouses merely *rush* the season a little. But in dark rooms using artificial light we can grow any plant we want at any time providing we supply the proper air conditions.

Mr. Boyd asked about the day-night cycle. We have made such experiments and found we could grow plants twice as fast by setting up a false day-night cycle, the plant taking on its food four hours under light and then growing eight hours in the dark. Sure enough, the plant did grow twice as fast. In other words, plants do not need the rest that human beings apparently require. Plants used were temperate climate plants and if you will think over our temperate climate with its storms, cloudiness and seasons you will probably reason why nature set up our plants to grow that way.

Mr. Boyd asked about the increase in efficiency and how it was indicated, and Mr. Locklin asked how it was measured. It is indicated by larger plants, more fruit, and more reproduction than normal. The measurement is purely a question of knowing how much sunlight hits the total area of the plant and how much of that goes into plant growth. This, of course, requires complete data on what happens to the plant and any parts of the plant from seed to harvest.

Mr. Miller's comment on special glass for greenhouses is acknowledged as a fine comment. We have done work on this and found that while it is a good answer at peak loads, it has some of the same limitations as shading at loads under peak. This was stated very well in other words by Mr. Flink.

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No. 1398

THERMAL CONDUCTIVITY OF SOILS FOR DESIGN OF HEAT PUMP INSTALLATIONS

By G. S. Smith*, Seattle, Wash., and Thomas Yamauchi**, Detroit, Mich.

THE DESIGN of ground grids as a source of heat for the heat pump demands a considerable knowledge of the thermal properties of the soil, the most important of which is thermal conductivity. A search through the handbooks will reveal a rather limited amount of information, usually of little value since the specific conditions are seldom given. Thermal studies of the soil around underground oil lines and electrical wire ducts, or for foundation design are more informative, though the intent of such research often results in information not well adapted for use in design of pipe grids in soil.

At the present time, workers in various parts of the country are making thermal conductivity studies on soils, and for conditions in their location. While the data obtained by each worker may be somewhat influenced by the difficulties involved in the method of study chosen, or by the peculiarities of his soils, nevertheless, there is little doubt that the outcome of the combined efforts of all will result in some very definite and dependable information of considerable value in the heating field, as well as in other fields. This report presents the results of some rather extensive studies on a few of the usual types of soils, under rather widely varying conditions. It represents only a small portion of a somewhat extensive research program on ground grids and the various factors involved in the use of heat pumps.

Problems Involved in Soil Thermal Conductivity Studies

The thermal conductivity of a soil varies greatly with its density and, to a lesser extent, with its moisture content. A change in the mean temperature of the soil causes only small changes in its conductivity, usually a small increase as the temperature increases. However, at temperatures well below freezing, the conductivity may again increase gradually, at least when some moisture is present. The nature of the soil is also a factor. The data presented seem to

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Presented at the Semi-Annual Meeting of The American Society of Heating and Ventilating Engineers, Muskoka, Ontario, Canada, June 1950.

indicate that the inherent density, or maximum density to which the soil could be reduced by eliminating all voids, would determine its maximum intrinsic heat conductivity.

Most laboratory tests involve a disturbance of the natural state of the soil in the preparations for such work and therefore, questions arise as to the similarity between factors derived from laboratory tests and those for the undisturbed soil. In an attempt to eliminate the possibility of any serious error of this type, three very different methods were used on certain samples, one of which

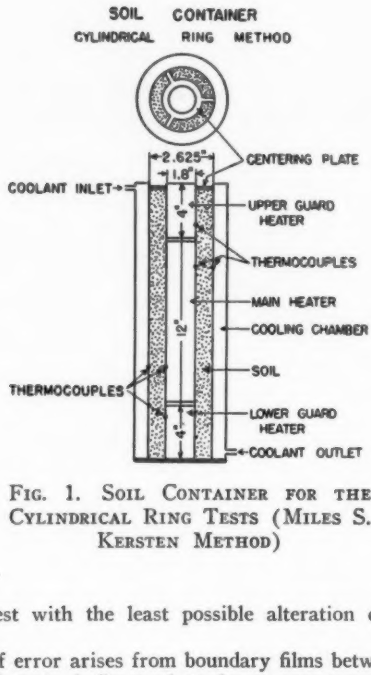

FIG. 1. SOIL CONTAINER FOR THE CYLINDRICAL RING TESTS (MILES S. KERSTEN METHOD)

accomplishes the test with the least possible alteration of the soil from its natural state.

Another source of error arises from boundary films between the soil and test equipment. Careful tests indicate that the temperature distribution at the boundary surfaces often differs somewhat from that in the main soil body. This problem has been studied by means of special tests where thermocouples were placed at various distances near such surfaces.

APPARATUS AND TEST PROCEDURE

First Method: The test equipment used in the first method described (the major portion of the data applies to this method) is similar to that used by Miles S. Kersten[1] at the University of Minnesota on tests conducted for the

[1] Exponent numerals refer to Bibliography.

U. S. Army. Fig. 1 shows the general construction features. The main and guard heaters are made of brass tubing with internal nichrome heaters. They are thermally insulated from each other by plastic rings 5/16 in. thick. The two outer tubes form a tempering jacket which is cooled by an alcohol-and-water mixture held at the desired temperature by an electric water cooler. Each heating section is controlled separately, and the power input to the main heater is measured by a wattmeter. All temperatures are measured by calibrated copper-

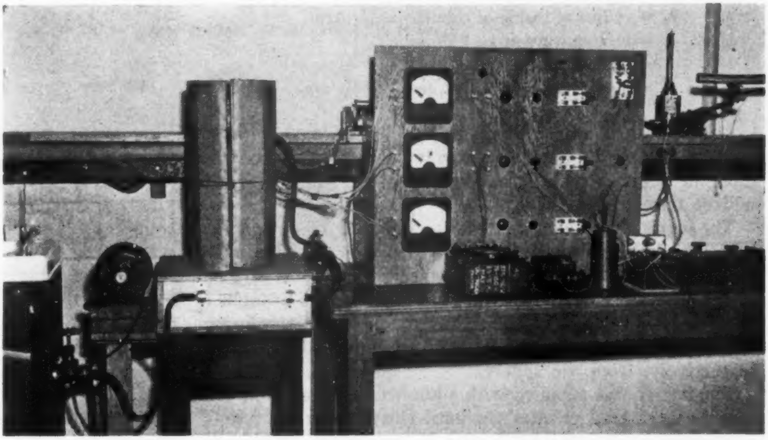

FIG. 2. VIEW OF APPARATUS USED ON TESTS

constantan thermocouples, using a potentiometer which can be read to a thousandth of a millivolt. The complete equipment used during the tests is shown in Fig. 2.

The procedure in making tests was as follows. With the volume of the test apparatus accurately known, a quantity of soil having the desired moisture content was carefully weighed out and divided into four parts. Then each fourth was packed sufficiently tight to fill one fourth of the soil space. The soil chamber was then sealed to prevent the loss of moisture.

The test was run by adjusting the three heaters and the cooler so that a difference in temperature of about 25 deg was maintained between hot and cold surfaces. The average of the hot and cold side temperatures was considered the average soil temperature. Also the temperatures of all heating sections were maintained equal within one degree or less. When these conditions were obtained and three readings taken at 10-min intervals gave results within 0.5 percent of each other, the test was considered complete. The duration of each test was three to four hours or longer.

The soil moisture was accurately measured as it was packed into the apparatus. Upon completion of the test and removal of the soil, a sample, taken from near the middle portion of the cylindrical ring, was again tested for moisture. If these two values did not check within one percent the test was discarded.

The following standard formula was used to determine conductivity:

$$k = \frac{q \log_e (r_2/r_1)}{2\pi\, l\, (t_1 - t_2)} \quad \text{(1)}$$

where

k = Btu per (hour) (square foot) (Fahrenheit degree per foot).
q = 3.414 × watts Btu per hour.
r_1 = radius of heating tube, feet.
r_2 = radius of inside of cooling jacket, feet.
t_1 = temperature at r_1, Fahrenheit.
t_2 = temperature at r_2, Fahrenheit.
l = length of main heater, feet.

Data obtained by this equipment will be termed *cylindrical ring tests.*

Second Method: In an attempt to compare results from laboratory tests to those in undisturbed soil, a few tests were made with a buried sphere. These will be termed *sphere tests.*

The equipment[2] for this work consisted of a spherical copper shell with heater and thermocouples as shown in Fig. 3. This sphere was placed at the bottom of a 5 ft deep hole drilled carefully and just large enough for the sphere to enter. All supply and thermocouple leads were brought to the surface through a ¾-in. plastic tube and the removed soil was carefully tamped back into place. Two other thermocouples were placed at horizontal distances of 20 and 27 in., respectively, from the sphere.

The heater was supplied with a known steady amount of power and all thermocouples were read at intervals until steady state conditions were obtained. This operation usually required several days. The thermal conductivity was then computed by the following formula.

$$k = \frac{q\,(r_2 - r_1)}{4\pi\,(t_1 - t_2)\,(r_1 r_2)} \quad \text{(2)}$$

All symbols are similar to those in Equation 1 except that r is now the distance from the center of the sphere to the thermocouple considered and t is the temperature measured by that couple.

Third Method: The third method employed is primarily a transient type of test to determine the thermal diffusivity of the soil, but the conductivity may be readily computed if the density and specific heat are also known. Details and theory of this method have been given by Carslaw[2], Newman[3], and Shannon and Wells[4], and will not be included here.

Fig. 4 shows the cylinder used for these tests. The container is packed full of soil at the density and moisture content desired, with one thermocouple on the surface of the cylinder and another fastened to plastic support at the geometric center of the soil within. The cylinder, sealed water tight, is immersed in a water bath kept at the desired temperature until both thermocouples indicate a uniform temperature throughout. It is then immersed in an ice bath and the temperature-time curve of the center couple is obtained by reading the temperature about every six minutes until equilibrium is reached. By means of this transient curve the diffusivity, and finally the thermal conductivity may be obtained.

For test purposes the specific heat was determined by calculating the volumetric heat capacity for the component parts in the equation

$$C = p\left(c + \frac{w}{100}\right) \quad \ldots \ldots \ldots \ldots \ldots \quad (3)$$

where

C = specific heat of sample, Btu per (cubic foot) (Fahrenheit degree).
c = specific heat of dry soil, Btu per (pound) (Fahrenheit degree).
p = dry density of the soil, pounds per cubic foot.
w = water content as percent of dry weight.

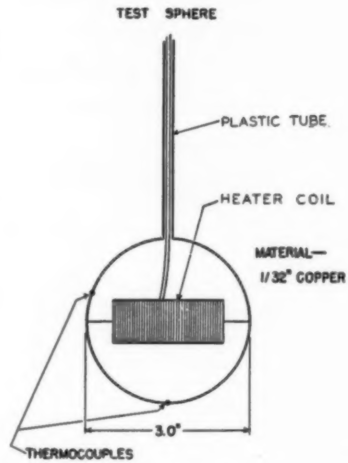

FIG. 3. APPARATUS USED FOR SPHERE TESTS

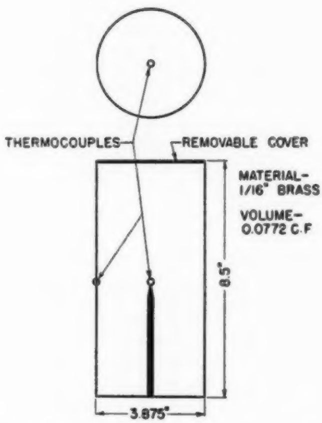

FIG. 4. CYLINDRICAL SOIL CONTAINER FOR TRANSIENT TYPE OF TEST

The specific heat of quartz sand was taken as 0.20 and for clay as 0.175. Water was taken as 1.0. Since the specific heat values of various types of soil vary only a small amount, the derived value appeared adequate for the comparison of results.

Soil Identification: To describe the soil samples, two of the more or less standard methods were used. The first, described in detail in Report of Subgrade Soils[5], identifies the soil by its liquid limit, plastic limit, and grain-size curve. The second is described by Casagrande[6] and identifies the sample according to the Casagrande table.

The liquid limit is obtained by forming a smooth layer of the soil and water mixture about three eighths of an inch thick in a suitable dish. When this is grooved by a grooving tool and tapped lightly against the palm of the hand ten times, if the lower soil edges just meet, the moisture content is termed the liquid limit.

The plastic limit is the moisture content of the soil at which it can be rolled into threads one-eighth inch in diameter without the thread breaking.

The Casagrande System grades soils in fifteen groups indicated by letters as follows:

Group	Compressibility and Expansion	Drainage Characteristics
GW	Almost none	Excellent
GC	Very slight	Practically impervious
GP	Almost none	Excellent
GF	Almost none to slight	Fair to practically impervious
SW	Almost none	Excellent
SC	Very slight	Practically impervious
SP	Almost none	Excellent
SF	Almost none to medium	Fair to practically impervious
ML	Slight to medium	Fair to poor
CL	Medium	Practically impervious
OL	Medium to high	Poor
MH	High	Fair to poor
CH	High	Practically impervious
OH	High	Practically impervious
Pt	Very high	Fair to poor

The letter designations are described as follows:

G — Gravels and gravelly soils
S — Sands and sandy soils
W — Well-graded, fairly clean
C — (When used as second letter). Well graded, excellent, clay binder
P — Poorly-graded or uniformly graded, fairly clean
F — Coarse materials containing fines, not covered by other groups
M — Inorganic silty and very fine sandy soils
C — (When used as first letter) Inorganic soils
O — Organic silts and clays
L — Fine-grained soils having liquid limit 50
H — Fine-grained soils having liquid limit 50
Pt — Fibrous organic (peat and swamp) soils

TABLE 1—SOIL CHART

Sample No.	Description	Sample Depth Ft	Undisturbed Density Lb Per Cu Ft	Plastic Limit	Liquid Limit	Casagrande Notation	Specific Gravity	Maximum Density Lb Per Cu Ft
2-M	Sand, gravel, some clay	2	90	......	......	S W	2.71	169
4-M	Sand, gravel, some clay	4	90	......	......	S W	2.71	169
6-M	Sand, gravel, some clay	6	90	......	......	S W	2.71	169
1-S	Clay	1	29.4	40.6	61	M.H.	2.64	165
2-5	Clay and loam	1	19.4	64.8	81.9	O.H.	2.39	149

The natural density of undisturbed soil of a sandy nature was obtained by digging a hole and obtaining the dry weight of the soil removed, then determining its volume by filling the hole with dry sand of known density. For wet clay or loam, a pipe, with the wall sharpened to a cutting edge on the outside

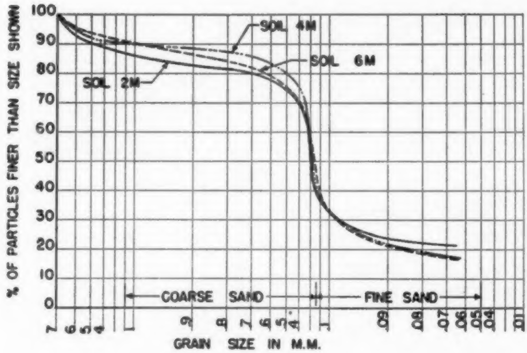

FIG. 5. GRAIN-SIZE CURVES FOR SANDY TYPE SOILS

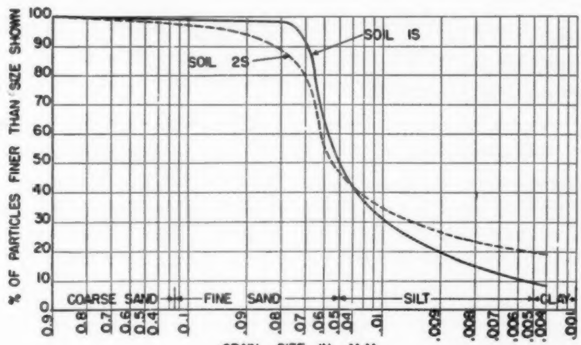

FIG. 6. GRAIN-SIZE CURVES FOR CLAY AND LOAM SOILS

periphery, was driven into the ground, and the depth of the hole and diameter of the pipe were carefully measured to determine volume.

All curves are plotted in terms of the dry density where the dry density is determined as follows:

$$\text{Dry density} = \text{Wet density} / (1 + \text{percent moisture}).$$

The percent moisture is based upon the dry weight of the sample. This procedure will allow better comparisons between various types of soil.

Results from Tests

Cylindrical Ring Method: Rather extensive tests were made on five samples of soil. Three of these were sandy in nature, a fourth was almost pure clay of a yellowish color, and a fifth, black loam containing some finely divided vegetable material. The soil chart (Table 1) gives the description and soil identification data on the five samples. Figs. 5 and 6 give the grain-size curves for the same samples.

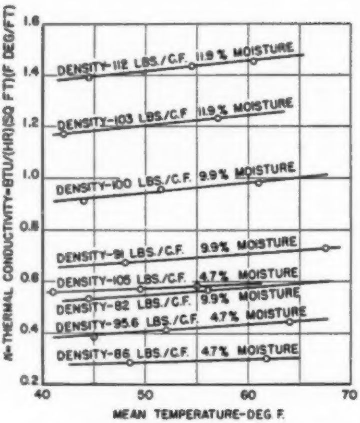

Fig. 7. Conductivity *vs.* Mean Temperature Curves for Soil Sample 6-M, at Various Dry Densities and Percentage Moisture Contents

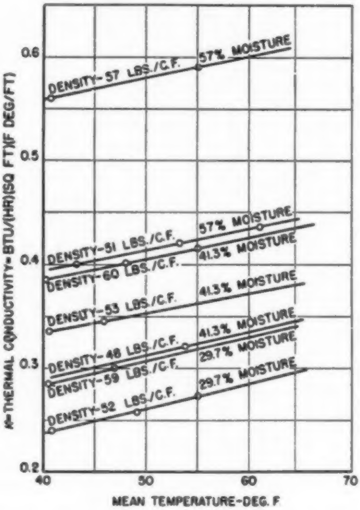

Fig. 8. Conductivity *vs.* Mean Temperature Curves for Soil 1-S at Various Dry Densities and Percentage Moisture Contents

Fig. 7 shows a series of curves for the 6-M sample which was quite sandy in nature. Here the thermal conductivity is plotted against mean temperature of the sample, each curve representing a definite density and moisture content. Fig. 8 presents similar curves for the clay type of soil. Actual test points are indicated. These curves indicate that the mean temperature of the soil has but little effect upon the thermal conductivity, though the density and moisture content are quite important. The relation of the conductivity to the mean temperature is quite linear over the temperature range used.

Replotting the data from curves as shown in Figs. 7 and 8, with conductivity on a log scale *vs.* dry density on a linear scale and all points selected at the same mean temperature, the results appear to be almost exactly linear. This same result was found to be true when plotting similar data reported by others. Assuming that the resulting curves should be straight lines, Fig. 9 presents

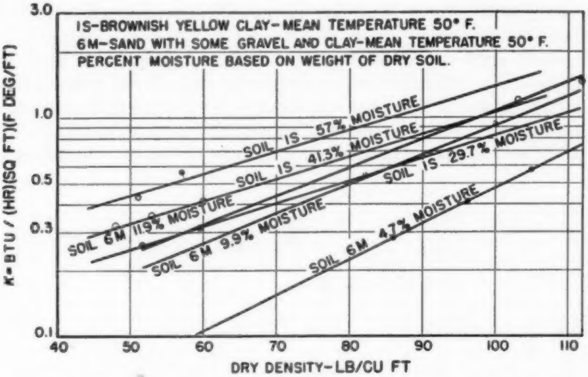

FIG. 9. CONDUCTIVITY *vs.* DRY DENSITY CURVES FOR SOILS 6-M AND 1-S AT 50 F MEAN TEMPERATURE AND FOR MOISTURE CONTENTS INDICATED

such curves for samples 6-M and 1-S. All curves are for a mean temperature of 50 F and each curve is for a given constant moisture content.

Perhaps the most useful method of presenting this information was obtained by replotting with percent moisture content on a log scale, and dry density on a linear scale. Again the mean temperature was maintained at 50 F, while each curve was plotted for a constant value of thermal conductivity. In some cases

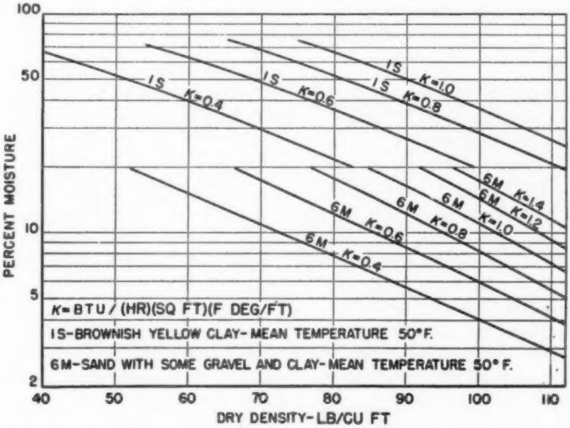

FIG. 10. PERCENT MOISTURE *vs.* DRY DENSITY CURVES FOR SOILS 6-M AND 1-S, AT 50 F MEAN TEMPERATURE AND FOR THERMAL CONDUCTIVITIES INDICATED

the resulting curves were straight lines. Only a few diverged far from straight lines.

Fig. 10 shows such curves for samples 6-M and 1-S, and Fig. 11, those for the 2-S sample. The other two sets of curves on Fig. 11 are replotted from work reported by Miles Kersten[1]. The set marked F. S. is for Fairbanks silt loam with an approximate analysis as follows: sand of various sizes, 7.6 percent; silt, 80.9 percent; and clay, 11.5 percent. This set of curves was taken at a mean

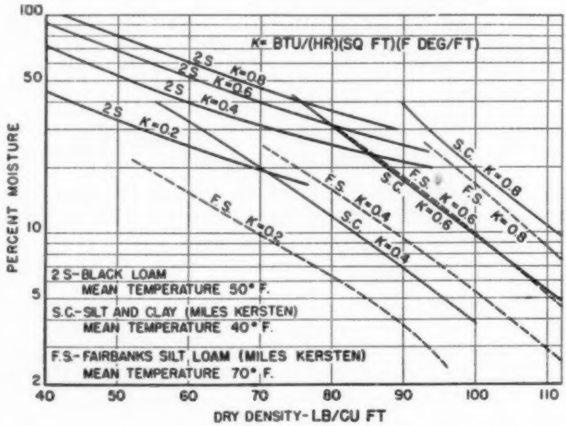

FIG. 11. PERCENT MOISTURE *vs.* DRY DENSITY CURVES FOR SOILS 2-S AND FOR SILT AND CLAY AND FAIRBANKS SILT FROM MILES KERSTEN REPORTS

temperature of 70 F. The set marked S.C. was described as silt and clay with, roughly, 30 percent sand, 15 percent clay, and the remainder silt. The mean temperature for these curves was 40 F.

The sandy type soils were all taken on the site chosen for experimental work with ground grids, and were so similar in thermal conductivity properties that only results from the sample 6-M taken at a six-foot depth are presented. The clay 1-S and black loam 2-S are definitely inferior heat conductors even at rather high moisture contents. This result was somewhat surprising since moisture would appear to be a very important factor, and clay types of soil will hold large percentages of moisture as compared to sandy soils. The superior heat conductivity of sand, together with the ability of clay to retain moisture, suggested tests on various mixtures of sand and clay to determine the possibility of some combination which might give a maximum conductivity.

For these tests, well graded sand was used of a size which passed through a No. 50 sieve, but was retained on a No. 100 sieve. The clay was of the blue-gray variety. The test made with pure sand contained 10 percent moisture which is less than the sand would hold. Other tests were made with various percentages of sand and clay until the last was of pure clay. The dry density

was held virtually constant on all tests, though the moisture content was allowed to increase in steps from 10 to 30 percent. Table 2 gives the results of the tests. Conductivity plotted against percent of clay is shown in Fig. 12.

Soil thermal conductivity thus appears to be greatly influenced by the specific density of the soil, and while moisture content aids, it is of lesser importance.

Effects of Surface Films and Moisture Migration: All of the tests so far mentioned were made by means of the cylindrical ring method. The major portion

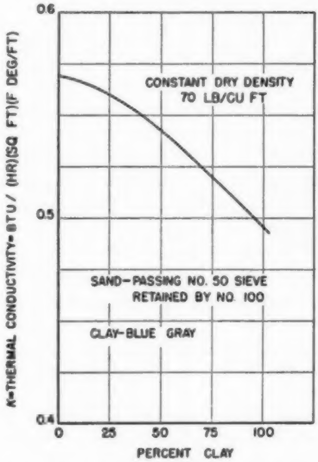

FIG. 12. CONDUCTIVITY OF SAND AND CLAY MIXTURES

was made by measuring only the temperatures at the hot and cold surfaces. To investigate the possibility of abnormal temperature changes due to surface film effects near the soil-metal junctions, a few tests were made by inserting extra thermocouples in the soil near both hot and cold surfaces. Thus, extra couples were held at ⅛- and ¼-in. distances from each surface by means of a light plastic strip. The results of these tests indicated a rather small deviation near the surfaces from a linear temperature distribution for sandy type of soil, with

TABLE 2—TESTS ON SAND-CLAY MIXTURES

SAND PERCENT	CLAY PERCENT	DRY DENSITY	MOISTURE PERCENT	k CONDUCTIVITY
100	0	70	10	0.570
75	25	70	15	0.560
50	50	70	20	0.545
25	75	70	25	0.520
0	100	70	30	0.495

a more pronounced deviation for clay, though not enough to make corrections desirable. Corrections indicated would result in thermal conductivities greater than for uncorrected values. Since the results of thermal conductivity tests cannot in general be duplicated with any great accuracy, and the lower values are on the safe side for design, the so-called uncorrected results have been given.

However, some interesting observations were indicated by these tests. The error was generally found to be greater on the heated side than on the cooled side. This difference is probably due to the tendency of the moisture to migrate from a heated, and toward a cooled surface. Thus grids used for space cooling must be more liberally designed than those used for space heating. A further study of this film effect for various pipe diameters, soil conditions, etc., seems very desirable.

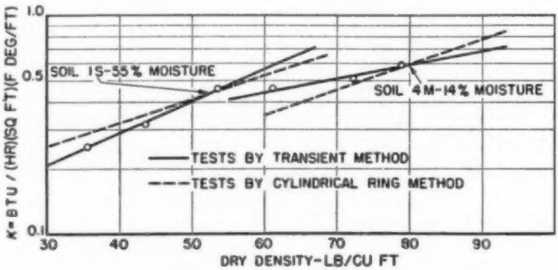

FIG. 13. CONDUCTIVITY *vs.* DRY DENSITY FOR TESTS ON SOILS 1-S AND 4-M FOR COMPARING CYLINDRICAL RING AND TRANSIENT METHODS

Comparison of Laboratory Tests and Undisturbed Soil Tests: A second test was made by the sphere method at a depth of about five feet below the surface and in the vicinity where the 6-M sample was obtained. This was necessarily at some little distance away. The undisturbed dry soil density was found to be about 90 lb per cu ft. At the time of the test the moisture content was 6 percent. The value of the thermal conductivity factor k from this test was found to be 0.565.

For comparison, the value of k for the same density and moisture from the curves determined by the cylindrical ring method was found to be 0.55. The comparison appears to be better than should be expected and later tests with the same sphere gave somewhat lower results but well within the possible percent of error. However, the results improve confidence in the cylindrical ring method as a means of thermal conductivity determination.

A second sphere was used to determine the conductivity of a very thick strata of hard pan soil which could not readily be tested by the cylindrical ring method. This strata was of a sand and clay type of soil containing almost no gravel and was only removable by means of a pick. The dry density was 138 lb per cu ft and moisture content at the time of the test was 8.3 percent. The thermal conductivity was 0.9 Btu per (hr) (sq ft) (F deg per ft).

Comparison by Transient Method: Several runs on samples 4-M and 1-S were carried out by the cylinder or transient method already described. The results as compared to those obtained by the cylindrical ring method are shown by curves in Fig. 13. Here again the results check reasonably well, though the theory and procedure differ greatly. This method proved to be the most convenient and speedy of the three methods used, and when only approximate results are necessary, would be quite adequate.

CONCLUSIONS

The measurement of thermal conductivity is a rather difficult process and even with extreme care will often give quite varied results. Comparison with the data from other workers is difficult because of possible differences in the nature of the soil. More standardization in methods of procedure and means of comparison and reporting is most desirable.

The material reported in this paper was obtained entirely from experimental determinations, and no attempt was made to derive general equations for mathematical determination. Miles Kersten[1] has demonstrated that, with sufficient experimental data, this can be done. Fig. 11, a composite chart, gives a comparison of results since the S.C. curve (silt-and-clay) was plotted from calculated data, while the 2-S and F-S were both from experimental results.

The thermal conductivity of any given soil is so greatly affected by the density that more attention should be given to the natural density of the undisturbed earth. Apparently the tendency is to pack soil in test apparatus to much greater densities than are found naturally. When soil is disturbed in placing ground grids, the replaced soil is probably less dense than before removal, and may require years to reach the same natural density. In design, every attempt should be made to use thermal conductivities applicable to the actual conditions. These conductivities may be much lower than might be expected.

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DISCUSSION

M. S. KERSTEN, Minneapolis: A knowledge of the thermal conductivity of soils is useful in many fields. In a recent study (Bibliography item 1) at the University of Minnesota for the Corps of Engineers, U. S. Army, this subject was investigated for

application to construction in regions of permafrost. This study included about 1,000 laboratory determinations of the thermal conductivity of various types of soils. The authors have made a valuable report and it is my intent to compare some of their results and conclusions with those of the study mentioned. The care used by the authors in presenting exact soil conditions (moisture content and dry density) for each test is to be commended since much of the earlier literature on soil conductivities is of meager value due to absence of such information. It is entirely possible that continued compilation of the test results of various investigators will permit development of charts by which reasonably accurate estimates of thermal conductivity can be made for a given set of soil conditions.

An attempt has been made, based on the University of Minnesota tests, to construct diagrams for estimating thermal conductivity values. Those shown in Fig. A are in

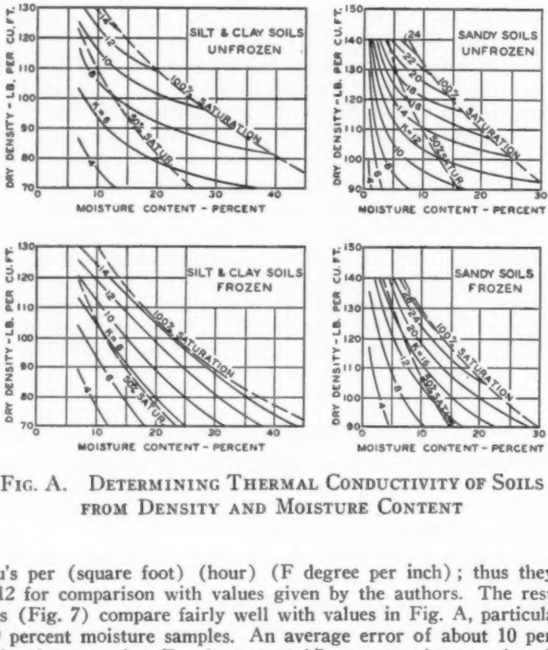

FIG. A. DETERMINING THERMAL CONDUCTIVITY OF SOILS FROM DENSITY AND MOISTURE CONTENT

units of Btu's per (square foot) (hour) (F degree per inch); thus they should be divided by 12 for comparison with values given by the authors. The results for the sand samples (Fig. 7) compare fairly well with values in Fig. A, particularly for the 9.9 and 11.9 percent moisture samples. An average error of about 10 percent would be obtained for these samples. For the tests at 4.7 percent moisture, values from Fig. A are much higher than the actual test results.

The tests on soil 1-S, as shown in Fig. 8, were made at moisture contents and densities outside the range of Fig. A. However, if use is made of the equation for silt and clay soils by which Fig. A was constructed and *k*-values are calculated for the moisture contents and densities given, an average error of about 17 percent is obtained, with a maximum error of 29 percent. From the information given, soils 1-S and 2-S must be highly organic soils. The undisturbed densities of 29.4 and 19.4 lb per cu ft are extremely low for socalled normal soils and are characteristic of soils such as peats. The densities at which tests were made (up to 60 lb per cu ft) are lower than is normally obtained in any earthwork in which topsoil is excluded. The test

results at high moisture contents (up to 57 percent) are extremely interesting since only a few of such tests have been reported elsewhere.

The statement of the authors that *The thermal conductivity of a soil varies greatly with its density and, to a lesser extent, with its moisture content* may be open to question. The comparison is difficult since the variables are not measured in the same units. A general rule concluded from the University of Minnesota studies was that a 1 lb increase in dry density resulted in about a 3 percent increase in thermal conductivity. A study of the slope of the curves in Figs. 9 and 13 would seem to verify this conclusion. Thus a 10-lb per cu ft increase in density would result in about a 30 percent increase in conductivity. Fig. 8 indicates that for soil 1-S, an increase in moisture content from 29.7 to 41.3 percent at a density of 59 or 60 lb per cu ft also increases the conductivity by about 30 percent. Also, in Fig. 7, a moisture change from 9.9 to 11.9 percent at densities of 100 and 103 lb per cu ft increases k by almost 30 percent. To say that the change in density is more important than the moisture change would be debatable. The importance of both items should be realized.

The experiment on the effect of mixing clay with sand in variable amounts and determining the effect on conductivity, as shown in Fig. 12, is interesting. It has been fairly well established that, at equal moisture contents and densities, the k-value is greater for coarse grained materials, such as sand and gravel, than for fine soils such as silt or clay. In the tests at the University of Minnesota, the highest k values obtained were for granular materials at relatively high densities and consequently relatively low moisture contents. For clay soils, as can be noted by a study of the silt and clay diagram, the k values for such soils at extremely high moistures are less than would be obtained at lower moistures but higher densities.

It is advisable in constructing charts such as Fig. 10 to limit the curves to areas which are possible conditions of moisture content and density. The 1-S, $k = 1.0$ and 1-S, $k = 0.8$ curves are almost entirely in an area of impossible conditions. For a soil with a specific gravity of 2.64, the maximum moisture contents possible at densities of 80, 90, 100 and 110 lb per cu ft are 40.2, 31.4, 24.5, and 18.9 percent, respectively.

L. R. INGERSOLL, Madison, Wis.: I have repeatedly used a scheme for soil diffusivity measurements somewhat similar to the authors' *third method,* with good effect. It has seemed better, however, to avoid the short cylinder container because of the uncertainty of thermal contact of the soil with the *top* surface, and to use a flat box container with one-dimensional heat flow.

The authors frankly point out the disadvantages of their main method as regards the necessary disturbance of the soil, but they indicate that this takes place likewise when a heat pump exchanger is installed. One would expect, however, that a year or two would cause relatively complete settling of the soil about an exchanger pipe grid, and the thermal conductivity would then be appreciably higher than that measured by their method. Accordingly it seems to me that more effort might well be expended on variations of their second method (which has been repeatedly used) which involves less soil disturbance. Transient state determinations might be utilized to shorten the time required for measurement.

AUTHOR'S CLOSURE (G. S. Smith): As has been pointed out by Professor Ingersoll, the *third method* of obtaining k by measuring the soil diffusivity is a very satisfactory means and requires very simply constructed apparatus. Perhaps it has the advantage of partly eliminating the film error at the junction between metal and soil as well as avoiding excessive moisture migration.

While the sphere method appears to offer little soil disturbance, the packing of the sphere in a deep hole little larger than the sphere offers possibilities of very poor surface junction conditions, such that much of the measured heat might be conducted only in specific directions. A pure transient method depending only upon the changes in temperature, as measured by two thermo measuring units spaced a short distance

from each other, and with readings taken at two or more definite intervals, would appear to be the most promising. We have attempted to use this method by employing the pulses or waves of heat travelling into the earth due to the sun's radiation on a warm day, but found the calculated *k* varied over a rather wide range and in general appeared too high. Perhaps greater precision than we were able to use might result in more satisfactory results.

Professor Ingersoll suggests that the soil conditions about the grid tube should approach the original soil conditions in a year or two. In my opinion it would require more time and in fact might never become normal if the soil adjacent to the tube is alternately frozen and thawed. Most economical grid designs would require some freezing during the critical portion of the heating season.

Professor Kersten's remarks are very interesting and point out certain suggestions we should have observed.

The 2-S soil was taken from a rather boggy spot and contained considerable fine vegetable matter. The 1-S soil contained little vegetable matter but was a very light clay. Both samples held large moisture content without appearing to be muddy.

The effect of moisture content, or the conductivity of the soil is a question which appears to deserve more study. *The Handbook of Chemistry and Physics** gives the *k* for sandstone as three to four times that for water and gypsum more than twice the water value. The presence of moisture in sand is probably more important as a means of filling the voids and improving material continuity than as a parallel path of better conductivity. When deprived of moisture some clays seem to be highly cellular and have qualities approaching those of a heat insulator. Thus the dry density of the soil appears to be a rather important factor.

Professor Kersten's equations for calculating the *k* for a soil are very desirable and more work might well be done in improving such methods.

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* Published by the Chemical Rubber Co., Cleveland, Ohio.

1399

No. 1399

HEAT FLOW THROUGH UNSHADED GLASS

Design Data for Use in Load Calculations

By G. V. PARMELEE* AND W. W. AUBELE**, CLEVELAND, OHIO

THE PURPOSE of this research paper is to present data on heat flow through glass sections that will be useful in the determination of loads on heating systems in winter and cooling systems in summer. The data are based principally upon the results of a long-range research program which has been carried on at the A.S.H.V.E. Research Laboratory in Cleveland under the direction of the Committee on Research and the Technical Advisory Committee on Heat Flow Through Glass.† These studies are a continuation of the pioneer investigations which were carried on at the Society's former Laboratory in Pittsburgh and have been reported in TRANSACTIONS, 1930 to 1941. Many of the basic data have already been given in recent A.S.H.V.E. research reports.[1, 2, 3, 4]

The data given in this report include: (1) overall coefficients of heat transmission for steady state heat flow through flat glass and glass block under winter conditions in the absence of solar radiation and (2) periodic heat flow through sunlit flat glass and glass block for four orientations at 40 deg north latitude on August 1.

DESIGN DATA

Design Conditions: The rate of heat flow through a glass panel is dependent upon the characteristics of a three-part system which consists of the glass section, the indoor environment, and the outdoor environment. The research carried on to date has resulted in the determination of basic thermal characteristics of glass sheets, assemblies of sheets, and walls of hollow glass block and mortar.

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[1] Exponent numerals refer to References.

Presented at the Semi-Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Muskoka, Ontario, Canada, June 1950.

Thermal characteristics here refer to the ability to conduct heat from one surface to another, the ability to transmit, absorb, reflect, and emit radiant energy and the ability to store heat. Geometrical characteristics such as surface roughness, size and installation details have a bearing on thermal behavior.

TABLE 1—COEFFICIENTS OF HEAT TRANSMISSION, *U*, OF GLASS SECTIONS

Coefficients are expressed in Btu per (hour) (square foot) (Fahrenheit degree difference in temperature between the air on the two sides), and are based on the following outdoor conditions: 0 F air temperature, clear skies, no solar radiation and a convection conductance of 4.0 Btu per (hr) (sq ft) (F deg difference in temperature)

SECTION A—FLAT GLASS

Number of Sheets	One	Two			Three		
Air Space, In.	....	¼	½	1	¼	½	1
U, Vertical	1.17	0.61	0.55	0.53	0.41	0.36	0.34
U, Horizontal Heat Flow Up	1.40	0.70	0.66	0.63	....	....	....

SECTION B—WALLS OF HOLLOW GLASS BLOCK[a]

Description[b]	Size of Block, In.	*U*
Types I and II	7¾×7¾×3⅞	0.55
Types I and II	5¾×5¾×3⅞	0.60
Types IV, IVA[c], V	7¾×7¾×3⅞	0.57
Type III	7¾×7¾×3⅞	0.48

[a] Values include mortar joint.
[b] See Table 2.
[c] Type IVA is the same as Type IV except external corrugations are vertical.

SECTION C— APPLICATION FACTORS FOR WINDOWS
MULTIPLY FLAT GLASS *U* VALUES BY THESE FACTORS

Window Description	Single Glass		Double Glass[d]		Windows with Storm Sash	
	Percent Glass	Factor	Percent Glass	Factor	Percent Glass	Factor
Sheets	100	1.00	100	1.00		
Wood Sash	80	0.90	80	0.95	80	0.90
Wood Sash	60	0.80	60	0.85	60	0.80
Metal Sash	80	1.00	80	1.20	80	1.00[e]

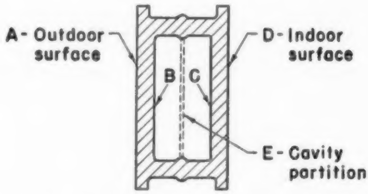
[d] Unit type double glazing, two lights or panes in same sash.
[e] Metal storm sash or auxiliary glass pane indoors.

The setting up of design data presents the problem of determining what should be the design conditions for the indoor and outdoor environments. In spite of much work done in the analysis of weather data, outdoor design conditions are at present inadequately described with regard to their coincidental effect on heat flow at outdoor surfaces. Design conditions indoors are likewise incom-

pletely described, because a statement of the indoor air temperature and humidity to be maintained does not specify the heat exchange between the room and the indoor surface nor the thermal storage characteristics of the structure. Furthermore, for the same season and location, design conditions are different for structures of different types and uses.

Hence, in the absence of more specific data on boundary heat transfer characteristics for design conditions, it has been necessary to select a combination of factors that might be expected to apply in many practical cases. Because methods of relating heating or cooling *loads* to instantaneous rates of heat flow

TABLE 2—DESCRIPTION OF GLASS BLOCK PATTERNS



Elevation Section of Hollow Glass Block to Indicate Location of Surface Patterns

Type I —Smooth Face
A, D: Smooth
B: Wide vertical ribs or flutes
C: Wide horizontal ribs or flutes
E: None

Type II —Semi-Light Diffusing
A, D: Narrow vertical ribs or flutes
B, C: Etched or stippled
E: None

Type III—Light Diffusing
A, D: Narrow vertical ribs or flutes
B, C: Etched or stippled
E: Glass fiber screen

Type IV—Light Diffusing
A, D: Close pitch deep horizontal corrugations
B, C: Vertical light diffusing prisms
E: None

Type V —Light Directing
A, D: Close pitch deep vertical corrugations
B, C: Horizontal light directing prisms
E: None

are not yet fully developed, the tables of data presented herewith give only *instantaneous rates of heat flow.*

Overall Coefficients: Design values of the overall coefficient of heat transmission, U, in Btu per (hour) (square foot) (Fahrenheit degree difference between indoor and outdoor air temperatures) have been computed for flat glass and hollow glass block and are given in Table 1. The glass block patterns are described in Table 2. The design conditions are based upon experimental work carried on by the A.S.H.V.E. Research Laboratory and by others[5, 6] and are described in detail in Appendix A. No serious error will be involved if the values are used for air temperatures within 10 to 15 deg of zero. The method of computing the values is given in Appendices B and C.

The data in Section A of Table 1 are applicable to glass sheets only. Section C gives approximate factors by which the Section A values are multiplied to obtain values for windows. The area to be used is the projected area of the glass and exposed sash.

Later in this report the coefficients given in Table 1 are compared with test results, and values for other conditions are presented.

TABLE 3—DESCRIPTION OF FLAT GLASS

GLASS NO.	FRACTION OF NORMALLY INCIDENT SOLAR RADIATION TRANSMITTED	COMMERCIAL TYPE GLASS
A	0.85—0.90	Ordinary window, High transmission plate
B	0.75—0.80	Regular plate
C	0.37—0.42	Heat absorbing
D	0.20—0.25	Heat absorbing

TABLE 4—INSTANTANEOUS RATES OF HEAT GAIN DUE TO TRANSMITTED DIRECT AND DIFFUSE SOLAR RADIATION BY VERTICAL UNSHADED FLAT GLASS

For clear atmospheres (see Table 12) on August 1 (18 deg declination, north) and 40 deg north latitude
For total instantaneous heat gain add these values to Table 5 values

INSTANTANEOUS HEAT GAIN FOR SINGLE SHEETS IN BTU PER (HR) (SQ FT)

Sun Time		Glass A				Glass B				Glass C				Glass D			
A.M.		N	E	S	W	N	E	S	W	N	E	S	W	N	E	S	W
5	7	3	7	1	0	2	6	0	0	1	3	0	0	1	2	0	0
6	6	24	133	8	7	20	114	7	6	10	61	4	4	5	32	2	2
7	5	13	198	14	12	11	168	12	10	6	91	7	6	3	47	4	3
8	4	12	211	22	15	10	177	18	13	6	94	11	7	3	48	6	4
9	3	13	186	48	18	11	154	39	15	6	81	21	9	4	42	11	5
10	2	14	133	76	20	12	110	62	16	7	56	32	10	4	28	16	5
11	1	14	64	96	21	12	52	79	18	7	31	40	10	4	14	20	6
12		14	22	104	22	12	18	85	18	7	11	44	11	4	6	22	6
	P.M.	N	W	S	E	N	W	S	E	N	W	S	E	N	W	S	E

INSTANTANEOUS HEAT GAIN FOR TWO SHEETS[a] IN BTU PER (HR) (SQ FT)

Sun Time		Glass A+A				Glass B+B				Glass C+B[b]				Glass C+C			
A.M.		N	E	S	W	N	E	S	W	N	E	S	W	N	E	S	W
5	7	2	6	0	0	2	4	0	0	1	3	0	0	1	2	0	0
6	6	19	120	7	6	13	90	6	5	7	52	3	3	5	36	3	2
7	5	11	177	13	11	8	133	10	8	5	76	6	5	3	53	4	4
8	4	11	188	18	13	8	139	14	10	5	80	8	6	4	55	6	5
9	3	12	165	36	15	9	120	26	12	5	69	15	7	4	46	10	5
10	2	12	115	62	17	10	83	44	13	5	47	24	8	4	30	16	6
11	1	13	51	81	18	10	35	57	14	6	20	32	8	4	13	20	6
12		13	19	88	19	10	15	62	15	6	9	35	9	4	7	22	7
	P.M.	N	W	S	E	N	W	S	E	N	W	S	E	N	W	S	E

[a] Spaced at ¼ in.
[b] Glass C is outdoor sheet.

TABLE 5—INSTANTANEOUS RATES OF HEAT GAIN BY CONVECTION AND RADIATION FROM VERTICAL UNSHADED FLAT GLASS

For clear atmospheres (see Table 12) on August 1 (18 deg declination, north) and 40 deg north latitude
For 75 F Indoor Temperature

For total instantaneous heat gain add these values to Table 4 values

INSTANTANEOUS HEAT GAIN FOR SINGLE SHEETS IN BTU PER (HR) (SQ FT)

Sun Time	Dry Bulb	Glass A				Glass B				Glass C				Glass D			
	F	N	E	S	W	N	E	S	W	N	E	S	W	N	E	S	W
5 a.m.	74	−1	−1	−1	−1	−1	−1	−1	−1	−1	−1	−1	−1	−1	−1	−1	−1
6	74	−1	0	−1	−1	1	4	−1	−1	4	15	0	0	5	21	1	1
7	75	0	2	0	0	1	9	1	1	3	30	2	2	4	43	3	3
8	77	2	3	2	2	3	13	3	3	4	40	4	5	5	57	5	6
9	80	5	6	6	5	6	17	7	6	7	47	11	8	8	65	12	9
10	83	9	10	10	9	10	20	12	10	11	45	20	12	12	57	23	13
11	87	13	14	14	13	14	21	19	14	15	34	31	16	16	39	36	17
12	90	17	17	18	17	18	20	24	18	20	22	40	20	21	24	46	22
1 p.m.	93	20	20	21	21	21	21	29	23	23	24	45	27	24	25	52	30
2	94	22	22	22	23	23	23	30	28	25	25	44	40	26	26	51	47
3	95	23	23	23	24	24	24	29	33	25	26	37	52	26	27	40	64
4	94	22	22	22	23	23	23	24	34	24	25	28	60	25	26	30	76
5	93	20	20	20	22	21	21	21	32	22	23	23	61	23	24	24	78
6	91	18	17	18	19	20	18	18	27	23	19	20	50	25	20	20	63
7	87	13	13	13	13	13	13	13	14	14	13	13	15	14	13	13	16
8	85	11	11	11	11	11	11	11	11	11	11	11	11	11	11	11	11
9	83	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8

INSTANTANEOUS HEAT GAIN FOR TWO SHEETS[a] IN BTU PER (HR) (SQ FT)

Sun Time	Dry Bulb	Glass A+A				Glass B+B				Glass C+B[b]				Glass C+C			
	F	N	E	S	W	N	E	S	W	N	E	S	W	N	E	S	W
5 a.m.	74	−1	−1	−1	−1	−1	−1	−1	−1	−1	−1	−1	−1	−1	−1	−1	−1
6	74	0	1	−1	−1	1	6	0	0	3	8	0	0	3	17	0	0
7	75	0	3	0	0	2	15	1	1	3	20	1	1	3	35	2	2
8	77	1	4	1	1	2	25	2	2	3	30	2	3	3	45	5	4
9	80	3	6	4	3	4	31	4	4	5	36	6	5	5	49	9	6
10	83	5	7	6	5	6	30	8	6	7	34	12	7	7	42	15	8
11	87	7	9	9	7	8	24	16	8	10	27	20	9	10	30	25	10
12	90	10	10	11	9	11	12	22	10	12	14	26	11	12	15	32	12
1 p.m.	93	12	12	13	12	13	13	26	13	14	14	30	15	14	15	36	19
2	94	12	12	14	15	13	14	28	20	15	15	31	22	15	16	35	30
3	95	13	13	14	16	14	15	25	27	15	15	27	33	15	16	30	47
4	94	12	12	13	16	13	14	19	34	14	14	20	40	14	15	23	57
5	93	12	12	12	15	13	13	14	36	13	13	15	44	13	14	16	58
6	91	11	10	10	12	13	11	11	28	14	11	11	36	14	12	12	45
7	87	7	7	7	7	7	7	7	9	8	7	7	9	8	7	7	10
8	85	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6
9	83	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5

[a] Spaced at ¼ in.
[b] Glass C is outdoor sheet.

Heat Flow Through Sunlit Glass: Instantaneous rates of heat gain due to transmitted direct and diffuse solar radiation for flat glass types, described in Table 3, are listed in Tables 4 and 6. Instantaneous rates of heat gain due to temperature difference and absorbed solar radiation are listed separately in

TABLE 6—INSTANTANEOUS RATES OF HEAT GAIN DUE TO TRANSMITTED DIRECT AND DIFFUSE SOLAR RADIATION BY VERTICAL UNSHADED FLAT GLASS

For industrial atmospheres (see Table 13) on August 1 (18 deg declination, north) and 40 deg north latitude

For total instantaneous heat gain add these values to Table 7 values

INSTANTANEOUS HEAT GAIN IN BTU PER (HR) (SQ FT)

Sun Time		Glass A				Glass B				Glass C				Glass D			
A.M.		N	E	S	W	N	E	S	W	N	E	S	W	N	E	S	W
5	7	2	5	0	1	1	4	0	1	1	2	0	0	0	1	1	0
6	6	15	75	8	6	14	63	7	5	7	35	4	3	3	18	2	2
7	5	14	135	16	12	12	115	13	10	7	63	8	6	4	33	4	3
8	4	16	149	24	17	13	125	20	14	8	68	12	8	4	35	6	5
9	3	17	138	43	21	15	115	35	17	8	61	19	10	5	32	10	6
10	2	18	104	63	24	15	85	52	20	9	49	28	12	5	23	14	6
11	1	18	57	78	26	15	47	64	22	9	28	34	13	5	13	18	7
12		18	29	84	29	15	25	69	25	9	14	36	14	5	8	19	8
	P.M.	N	W	S	E	N	W	S	E	N	W	S	E	N	W	S	E

INSTANTANEOUS HEAT GAIN FOR TWO SHEETS[a] IN BTU PER (HR) (SQ FT)

Sun Time		Glass A+A				Glass B+B				Glass C+B[b]				Glass C+C			
A.M.		N	E	S	W	N	E	S	W	N	E	S	W	N	E	S	W
5	7	2	5	0	1	1	3	0	1	1	2	0	0	0	1	0	0
6	6	12	67	7	5	8	50	6	4	5	28	3	2	3	21	3	2
7	5	12	121	14	10	9	90	11	8	5	52	6	5	4	37	5	4
8	4	14	133	21	15	11	99	16	12	6	57	9	7	5	40	7	5
9	3	15	123	34	18	12	90	25	14	7	52	14	8	5	36	10	6
10	2	15	90	52	21	12	65	38	16	7	37	21	9	5	25	14	7
11	1	16	46	67	23	12	33	48	18	7	19	27	10	5	13	18	8
12		16	26	72	26	12	20	52	20	7	11	29	11	5	9	19	9
	P.M.	N	W	S	E	N	W	S	E	N	W	S	E	N	W	S	E

[a] Spaced at ¼ in.

[b] Glass C is outdoor sheet.

Tables 5 and 7. Tables 8, 9, 10 and 11 give similar data for walls of eight-inch square hollow glass block. These tables are for 40 deg north latitude on August 1 (18 deg declination, north). In each case the total heat flow is the sum of the transmitted solar radiation and the convection and radiation gain, that is, Table 4 values are added to Table 5 values and so on.

TABLE 7—INSTANTANEOUS RATES OF HEAT GAIN BY CONVECTION AND RADIATION FROM VERTICAL UNSHADED FLAT GLASS

For industrial atmospheres (see Table 13) on August 1 (18 deg declination, north) and 40 deg north latitude

For 75 F Indoor Temperature

For total instantaneous heat gain add these values to Table 6 values

INSTANTANEOUS HEAT GAIN FOR SINGLE SHEETS IN BTU PER (HR) (SQ FT)

Sun Time	Dry Bulb	Glass A				Glass B				Glass C				Glass D			
	F	N	E	S	W	N	E	S	W	N	E	S	W	N	E	S	W
5 a.m.	74	−1	−1	−1	−1	−1	−1	−1	−1	−1	−1	−1	−1	−1	−1	−1	−1
6	74	−1	−1	−1	−1	0	1	−1	−1	2	7	0	0	3	10	1	0
7	75	0	1	0	0	1	5	1	1	3	20	2	2	5	28	3	2
8	77	2	2	2	2	3	10	3	3	5	30	5	5	6	40	6	6
9	80	5	6	6	5	6	15	7	6	8	36	9	9	9	48	11	10
10	83	9	9	9	9	10	18	12	10	12	36	17	13	13	46	21	14
11	87	13	14	14	13	14	19	18	15	16	31	27	17	17	37	32	18
12	90	17	17	17	17	18	19	22	18	20	24	34	21	21	26	40	23
1 p.m.	93	20	20	21	21	21	22	26	22	23	25	39	26	25	27	45	28
2	94	22	22	22	23	23	23	27	26	25	26	40	35	27	27	45	41
3	95	23	23	23	24	24	24	27	29	26	26	36	45	27	28	38	53
4	94	22	22	22	23	23	23	24	30	24	25	29	48	26	26	31	60
5	93	20	20	20	21	21	21	21	29	22	23	24	47	23	23	24	59
6	91	18	18	18	18	19	18	18	23	21	19	19	35	22	19	19	43
7	87	13	13	13	13	13	13	13	13	14	13	13	14	14	13	13	15
8	85	11	11	11	11	11	11	11	11	11	11	11	11	11	11	11	11
9	83	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8

INSTANTANEOUS HEAT GAIN FOR TWO SHEETS[a] IN BTU PER (HR) (SQ FT)

Sun Time	Dry Bulb	Glass A+A				Glass B+B				Glass C+B[b]				Glass C+C			
	F	N	E	S	W	N	E	S	W	N	E	S	W	N	E	S	W
5 a.m.	74	−1	−1	−1	−1	−1	−1	−1	−1	−1	−1	−1	−1	−1	−1	−1	−1
6	74	−1	0	−1	−1	1	2	0	0	1	4	0	0	2	9	0	0
7	75	0	1	0	0	2	10	1	1	2	13	2	1	3	22	2	1
8	77	1	2	1	1	2	17	2	3	3	21	3	3	4	31	3	4
9	80	3	4	4	4	4	23	4	4	5	27	6	5	6	35	8	6
10	83	5	6	6	5	6	24	8	7	7	28	10	8	8	33	14	9
11	87	8	9	9	8	9	21	14	9	10	24	16	9	11	26	21	11
12	90	10	10	11	10	11	14	20	11	12	16	22	12	13	17	27	14
1 p.m.	93	12	12	13	13	13	14	24	14	14	15	26	15	15	16	31	18
2	94	13	13	14	14	14	14	24	16	15	16	27	19	15	17	31	25
3	95	13	13	14	15	15	15	23	23	15	16	24	27	16	17	27	36
4	94	13	13	13	15	14	14	19	26	14	15	14	31	15	16	21	42
5	93	12	12	12	14	12	12	13	25	13	13	14	30	13	14	15	42
6	91	10	10	10	11	12	10	12	22	13	11	12	25	13	11	13	29
7	87	7	7	8	7	7	7	9	11	8	7	9	11	8	7	10	11
8	85	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6	6
9	83	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5

[a] Spaced at ¼ in.

[b] Glass C is outdoor sheet.

TABLE 8—INSTANTANEOUS RATES OF HEAT GAIN DUE TO TRANSMITTED DIRECT AND DIFFUSE SOLAR RADIATION BY WALLS OF HOLLOW GLASS BLOCK

For clear atmospheres (see Table 12) on August 1 (18 deg declination, north) and 40 deg north latitude

For total instantaneous heat gain add these values to Table 9 values

INSTANTANEOUS HEAT GAIN IN BTU PER (HR) (SQ FT)

Sun Time		Glass Block No. I				Glass Block No. II				Glass Block No. III				Glass Block No. IV			Glass Block No. IVA			Glass Block No. V		
A.M.	P.M.	N	E	S	W	N	E	S	W	N	E	S	W	E	S	W	E	S	W	E	S	W
5	7	0	3	0	0	0	3	0	0	0	2	0	0	1	0	0	1	0	0	1	0	0
6	6	4	66	3	2	4	64	3	2	3	44	2	2	24	2	1	24	2	1	22	2	1
7	5	3	97	5	4	3	89	5	4	3	58	3	3	33	3	2	39	3	2	37	3	2
8	4	4	90	6	5	4	78	6	5	3	52	5	4	26	4	3	39	4	3	58	4	3
9	3	4	59	11	6	4	50	10	6	3	37	7	4	16	6	3	37	5	3	53	6	3
10	2	5	28	16	6	5	28	15	6	3	21	10	5	10	8	4	24	9	4	31	13	4
11	1	5	14	18	7	5	13	18	7	3	9	14	5	7	8	4	9	17	4	10	18	4
12		5	7	17	7	5	7	17	7	3	5	15	5	4	8	4	4	20	4	4	20	4
P.M.		N	W	S	E	N	W	S	E	N	W	S	E	W	S	E	W	S	E	W	S	E

TABLE 9—INSTANTANEOUS RATES OF HEAT GAIN BY CONVECTION AND RADIATION FROM SUNLIT WALLS OF HOLLOW GLASS BLOCK

For clear atmospheres (see Table 12) on August 1 (18 deg declination, north) and 40 deg north latitude

For 75 F Indoor Temperature

For total instantaneous heat gain add these values to Table 8 values

INSTANTANEOUS HEAT GAIN IN BTU PER (HR) (SQ FT)

Sun Time	Dry Bulb	Glass Block Nos. I, II				Glass Block No. III				Glass Block Nos. IV, IVA, V		
	F	N	E	S	W	N	E	S	W	E	S	W
5 a.m.	74	0	0	0	0	0	0	0	0	0	0	0
6	74	2	6	0	0	2	10	0	0	10	0	0
7	75	3	19	1	1	3	26	1	1	27	1	1
8	77	2	28	3	2	2	37	3	2	38	4	2
9	80	2	35	7	4	2	38	7	4	41	7	4
10	83	4	38	14	6	4	37	14	6	40	14	6
11	87	6	31	21	8	6	31	21	8	33	21	8
12	90	8	18	25	10	8	18	24	10	20	26	10
1 p.m.	93	10	16	28	14	10	16	27	14	17	28	13
2	94	11	17	27	30	11	17	26	29	18	28	28
3	95	13	18	23	43	13	18	22	41	19	25	41
4	94	13	17	18	46	13	17	17	50	18	19	51
5	93	13	15	15	44	13	15	15	53	17	15	55
6	91	15	13	13	37	15	13	13	45	15	13	49
7	87	13	10	10	23	13	10	10	27	10	10	30
8	85	9	6	6	9	9	6	6	9	6	7	11
9	83	9	4	4	6	6	4	4	6	4	4	7

TABLE 10—INSTANTANEOUS RATES OF HEAT GAIN DUE TO TRANSMITTED DIRECT AND DIFFUSE SOLAR RADIATION BY WALLS OF HOLLOW GLASS BLOCK

For industrial atmospheres (see Table 13) on August 1 (18 deg declination, north) and 40 deg north latitude

For total instantaneous heat gain add these values to Table 11 values

INSTANTANEOUS HEAT GAIN IN BTU PER (HR) (SQ FT)

Sun Time		Glass Block No. I				Glass Block No. II				Glass Block No. III				Glass Block No. IV			Glass Block No. IVA			Glass Block No. V		
A.M. ↓		N	E	S	W	N	E	S	W	N	E	S	W	E	S	W	E	S	W	E	S	W
5	7	0	3	0	0	0	2	0	0	0	2	0	0	1	0	0	1	0	0	1	0	0
6	6	3	35	3	2	3	34	3	2	2	23	2	1	13	2	1	13	2	1	13	2	1
7	5	4	63	5	4	4	59	5	4	3	38	4	3	23	3	2	27	3	2	25	3	2
8	4	5	61	7	6	5	55	7	6	4	36	5	4	20	4	3	27	4	3	39	4	3
9	3	6	44	11	7	6	38	10	7	4	29	7	5	14	6	4	28	6	4	37	7	4
10	2	6	24	15	8	6	24	14	8	4	18	10	6	10	8	4	19	9	4	23	11	5
11	1	6	14	17	9	6	14	17	9	4	10	13	6	7	8	5	9	14	5	10	15	5
12		6	10	17	10	6	10	17	10	4	7	14	7	5	8	5	5	16	5	6	16	6
	↑ P.M.	N	W	S	E	N	W	S	E	N	W	S	E	W	S	E	W	S	E	W	S	E

TABLE 11—INSTANTANEOUS RATES OF HEAT GAIN BY CONVECTION AND RADIATION FROM SUNLIT WALLS OF HOLLOW GLASS BLOCK

For industrial atmospheres (see Table 13) on August 1 (18 deg declination, north) and 40 deg north latitude

For 75 F Indoor Temperature

For total instantaneous heat gain add these values to Table 10 values

INSTANTANEOUS HEAT GAIN IN BTU PER (HR) (SQ FT)

Sun Time	Dry Bulb	Glass Block Nos. I, II				Glass Block No. III				Glass Block Nos. IV, IVA, V		
	F	N	E	S	W	N	E	S	W	E	S	W
5 a.m.	74	0	0	0	0	0	0	0	0	0	0	0
6	74	1	3	0	0	1	5	0	0	5	0	0
7	75	2	11	1	1	2	16	1	1	16	1	1
8	77	2	19	3	2	2	25	3	2	25	3	2
9	80	3	25	5	4	3	28	5	4	29	5	4
10	83	4	28	11	6	4	28	11	7	29	11	7
11	87	6	25	17	8	6	24	17	10	26	17	9
12	90	8	17	22	10	8	18	22	12	19	22	11
1 p.m.	93	10	17	25	13	10	17	25	14	18	25	13
2	94	12	18	25	23	12	18	25	24	19	25	23
3	95	13	19	21	33	13	19	22	33	19	23	32
4	94	14	18	17	36	14	18	18	38	18	18	40
5	93	13	17	15	35	13	17	16	38	17	16	41
6	91	14	14	13	28	14	14	13	30	14	14	35
7	87	11	10	10	18	11	10	10	17	10	11	22
8	85	8	6	7	9	8	6	7	9	6	7	10
9	83	5	4	4	6	5	4	4	6	4	4	6

TABLE 12—DESIGN VALUES OF SOLAR RADIATION RECEIVED BY VARIOUSLY ORIENTED SURFACES IN BTU PER (HR) (SQ FT) FOR CLEAR ATMOSPHERES

Solar Altitude Deg	Direct Radiation[a]	Diffuse[b] or Sky Solar Radiation Received by Walls and a Horizontal Surface				
A.M.	Normal to Sun	N	E	S	W	Hor.
5	67	4	9	4	4	7
10	123	7	18	9	8	14
15	166	10	24	13	11	19
20	197	12	28	16	13	23
25	218	13	31	18	15	26
30	235	14	33	21	17	28
35	248	15	34	23	18	30
40	258	15	34	25	20	31
45	266	16	34	26	21	32
50	273	16	33	28	23	33
60	283	17	31	29	25	34
70	289	17	26	30	26	35
P.M.	Normal to Sun	N	W	S	E	Hor.

a Moon's[7] proposed standard for sea level, 20 mm precipitable water vapor, 300 dust particles per cu cm., 2.8 mm Hg partial pressure of ozone.

b For 40 deg north latitude on about August 1.

Tables 4, 5, 8, and 9 may be considered as applicable to localities having relatively clear atmospheres, while Tables 6, 7, 10 and 11 are applicable to humid industrialized areas. Local conditions should be investigated fully before selecting a table for design use. The tabular values of Tables 5, 7, 9, and 11

TABLE 13—DESIGN VALUES OF SOLAR RADIATION RECEIVED BY VARIOUSLY ORIENTED SURFACES IN BTU PER (HR) (SQ FT) FOR INDUSTRIAL ATMOSPHERES

Solar Altitude Deg	Direct Radiation[a]	Diffuse[b] or Sky Solar Radiation Received by Walls and a Horizontal Surface				
A.M.	Normal to Sun	N	E	S	W	Hor.
5	34	4	11	5	3	9
10	58	8	22	9	7	18
15	80	11	28	13	9	24
20	103	13	36	17	12	31
25	121	16	43	21	16	38
30	136	18	47	24	18	44
35	148	19	50	27	21	48
40	158	20	50	30	23	52
45	165	21	49	31	25	55
50	172	22	47	34	27	58
60	181	22	41	37	30	63
70	188	22	34	41	34	69
P.M.	Normal to Sun	N	W	S	E	Hor.

a Derived from recommended design sol-air temperatures[9] for a horizontal surface with absorptivity of 1.0 for New York City.

b For 40 deg north latitude on about August 1.

are based upon an outdoor design dry bulb temperature of 95 F maximum and an indoor temperature of 75 F dry bulb. The solar intensity values for a clear atmosphere (Table 12) are based upon Moon's[7] suggested standard curve. This curve is practically identical with values derived from recommended design sol-air data[8] for Lincoln, Neb. The values for an industrial atmosphere (Table 13) were derived from design sol-air data[9] for New York City. The solar radiation data are discussed further in Appendix A. The calculation procedures followed in setting up the tables are discussed fully in Appendices A, B, and C and will be of interest to those who desire to work out values for other conditions and locations.

A surface conductance of 4.0 Btu per (hr) (sq ft) (F deg) has been used in computing heat transfer between the outdoor environment and the outdoor

TABLE 14—CORRECTIONS TO TABLES 5, 7, 9 AND 11 FOR INDOOR AND OUTDOOR TEMPERATURE

For each degree the design room temperature exceeds 75 F, subtract correction. For each degree the design outdoor dry bulb temperature exceeds 95 F, add correction. Apply the corrections to each value in the appropriate table

GLASS AREA	CORRECTION BTU (HR) (SQ FT)
Single Glass	1.0
Double Glass	0.6
Glass Block (Types I, II, IV, V)	0.5
Glass Block (Type III)	0.4

surface of glass panels. This includes convection and all radiation exchanges, and hence is an approximation. It is, however, consistent with present practice in estimating heat gain through sunlit walls and roofs.

DISCUSSION OF NEW DATA

Overall Coefficients: In Part I of A.S.H.V.E. RESEARCH BULLETIN No. 1, Heat Transmission through Glass,[10] the literature dealing with overall coefficients is reviewed. In the summary of this review, it was stated that differences in the results of investigators who tested what appeared to be similar or identical installations can be attributed to differences in the degree of circulation of air in the test enclosure, in the emissive character of the hot box surfaces, and in the mean temperature and air-to-air temperature difference.

These explanations, which are concerned with the characteristics of the indoor and outdoor environment, apply here.

Fig. 1 shows the influence of the outdoor environment on values of U for flat glass with the outdoor temperature at 0 F and the indoor conditions held constant. Also shown are curves for indoor and outdoor conditions comparable to those in tests made with the calorimeter apparatus and previously reported.[3] It should be noted that indoor surface conductances for the calorimeter, for combined radiation and convection, are of the order of 1.20 as compared to 1.40 to 1.50 Btu per (hr) (sq ft) (F deg) for design conditions. Test points are shown directed to the appropriate curves. These points were located along the abscissa in accordance with the convection conductances computed

from flat plate tests.[11] The point for double glass is high because the sky was not completely overcast. No measurements of radiation received from the outdoor surroundings were made in these tests.

The results of recent hot box tests, some of which were cited as reference material in an A.S.H.V.E. paper,[3] are in good agreement with the curves of Fig. 1 for overcast skies. The radiation characteristics of the out-of-doors with overcast skies are closely comparable to conditions in the hot box tests, since the hot box surfaces were covered with non-reflective materials. The values in

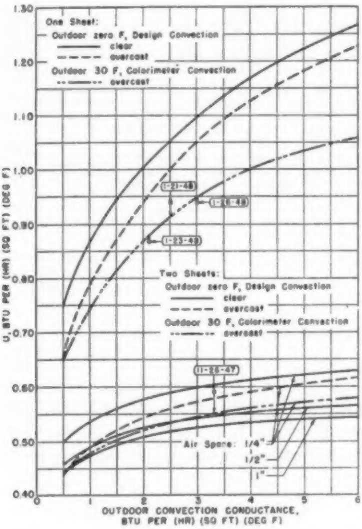


FIG. 1. INFLUENCE OF OUTDOOR ENVIRONMENT ON U VALUES FOR VERTICAL FLAT SHEETS OF GLASS AND COMPARISON WITH TEST RESULTS

Table 1 therefore appear to be reasonable and are recommended for use in place of the present values[12] of 1.13, 0.45, and 0.26 Btu per (hr) (sq ft) (F deg) for single, double and triple glass respectively.

The factors for windows were derived from data reported in Bulletin No. 1[10] by comparing U values for sheets of glass with U values for windows. Each comparison was made between tests performed in the same hot box under identical conditions.

The suggested coefficients for glass block, except for Type III, are 10 to 20 percent higher than published values[12] of 0.49 and 0.46 for smooth and ribbed faced block respectively. Reference 24 in Bulletin No. 1,[10] from which these values were derived, shows that both had U values of 0.43 when tested under identical conditions of low air velocity and low emissivity box surfaces. This

value was adjusted to design conditions of 15 mph by empirical formulas for surface conductances. Inspection of the data will show that the panel conductances, based on panel surface temperatures, were 1.00 and 1.05 Btu per (hr) (sq ft) (F deg) for the panels of smooth and ribbed faced block respectively. These compare favorably with 0.98 and 1.09 found in the calorimeter tests[4], on which the Table 1 values were based. The differences between the *U* values therefore are due to the characteristics of the indoor and outdoor environments.

Fig. 2 shows the influence of the outdoor environment on *U* values for glass block walls and compares test data with calculated curves. Except for the Type

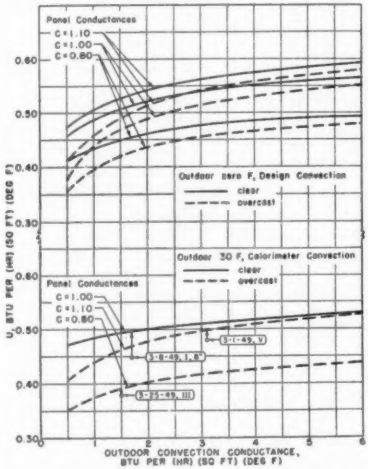


FIG. 2. INFLUENCE OF OUTDOOR ENVIRONMENT ON *U* VALUES FOR GLASS BLOCK WALLS AND COMPARISON WITH TEST RESULTS

I block, the points are in good agreement with the curves. In the test of this block the radiant energy contribution from the surroundings was 7 Btu per (hr) (sq ft) in excess of that used for design purposes. Allowance for this will place the test point within 2 percent of the curve.

Heat Flow Through Sunlit Flat Glass: For purposes of standardization in making up Tables 4, 5, 6, and 7, each set of values was computed for a glass of definite solar energy transmitting characteristics (see Appendix C). In practice the transmitting characteristics of a given type of glass are variable. Hence each set of values has been designated as being applicable to any glass that has a transmittance which falls within the range indicated in Table 3. Over such a range the sum of the transmittance and absorptance is approximately constant, so that the *total* heat flow is substantially constant. The division between gain from transmitted solar energy and that from convection and radiation gain is, however, altered.

It will be noted that glasses C and D, termed heat absorbing glasses, cause a marked reduction in the transmitted solar energy as compared with glasses A and B. This reduction is accompanied by a considerable increase in convection and radiation heat gain. As shown in Fig. 3 environmental conditions significantly affect the convection and radiation heat gain.

The convection and radiation gains for two sheets of glass are for one-quarter inch spacing. An analytical investigation was made to determine the effect of air space for the combination of glasses B and C at the period of peak heat gain. The results, shown in Fig. 4, indicate that air space widths between one-quarter

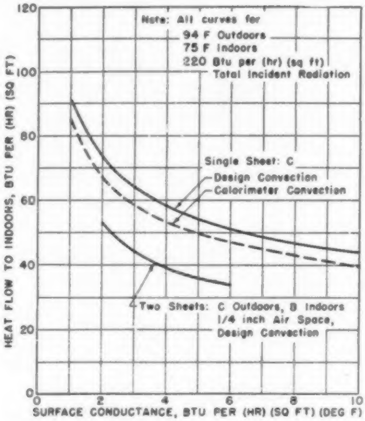


FIG. 3. INFLUENCE OF OUTDOOR SURFACE CONDUCTANCE ON CONVECTION AND RADIATION HEAT GAIN FROM SUNLIT GLASS

inch and one inch have no important influence. Spacing affects transmitted radiation only to the extent that the setback shades the inner sheet. This is significant at high angles of incidence.

Heat Gain Through Sunlit Glass Block: The heat gain data for sunlit glass block are principally new data. A.S.H.V.E. RESEARCH REPORT No. 1147[13] gives the results of an investigation of the heat flow characteristics of two of the types for which data are given in this paper, Type I and Type V. Data on a third type, similar to Type I, were also given. Design data were given for Type I for east, south and west walls and for Type V for south facing walls. It is values for the former that have been given in THE HEATING, VENTILATING, AIR CONDITIONING GUIDE 1941 to 1950. The data presented in this report therefore represent a considerable extension of earlier work.

APPLICATION OF DATA

Overall Coefficients: The values presented in Table 1 are applicable to most cases encountered in estimating heating loads. However, unusual conditions

of either air motion or radiant energy exchange or both should be examined carefully to determine whether or not they deviate greatly from the design conditions. Figs. 1 and 2 take into account most variations in outdoor conditions. Heat flow calculations for greenhouses and sun rooms and for rooms in which there is unusual air motion in the vicinity of the indoor glass surfaces must be based upon a careful analysis of the situation. In the absence of field data related to convection and radiation exchanges in such cases, no application factors can be given at this time.

Solar Heat Gain: In the past three years, a considerable amount of solar radiation data has been collected during the course of test work. While no correlation of the data with other weather conditions has yet been attempted, some comments of a general nature may be helpful in selecting the proper

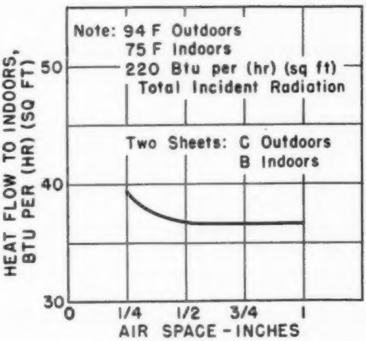


FIG. 4. INFLUENCE OF AIR SPACE WIDTH ON CONVECTION AND RADIATION HEAT GAIN FROM SUNLIT GLASS

radiation data to use for design purposes. It has been observed in Cleveland that solar intensity values of the order of those given for industrial atmospheres, are usually associated with dry bulb and wet bulb temperatures near the design values of 95 F and 75 F (67 F dewpoint). On the other hand, values equal to or exceeding the suggested values for a clear atmosphere are often encountered during Cleveland summers, but almost always with the dry bulb 10 to 15 deg below the design value and the dewpoint temperatures at about 55 F.

Therefore, although the values for humid industrial atmospheres will be applicable to many cities for summer *design conditions,* the possibility should not be overlooked that rooms having sun exposures may have peak loads on days when temperatures are well below design values. The total plant capacity may be entirely adequate, but the room distribution system may be grossly underdesigned if proper consideration is not given to sun loads under these conditions.

If it is desired to use Tables 5, 7, 9 and 11 for other indoor temperatures, approximate corrections can be made according to Table 14.

It is beyond the scope of this paper to treat the problem of translating instantaneous rates of heat flow into apparatus loads. Because both transmitted solar

energy and radiant energy from the indoor glass surfaces are absorbed by the indoor surfaces to become stored energy or to enter the air stream by convection from these surfaces, apparatus loads are lower than instantaneous rates of heat gain. The reader is referred to recent papers by C. O. Mackey[14] and C. S. Leopold[15, 16] for studies of this problem.

ACKNOWLEDGMENT

The authors acknowledge with thanks the suggestions and assistance given by the members of the Technical Advisory Committee on Heat Flow Through Glass and by their colleagues of the A.S.H.V.E. Research Laboratory staff in preparing this paper.

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17. See p. 395 of Reference 5.

18. See p. 268 of Reference 12.

19. Free Heat Convection Through Enclosed Plane Gas Layers, by Max Jakob (*A.S.M.E. Transactions,* April 1946, p. 189).

APPENDIX A

DESIGN CONDITIONS

Indoor Design Conditions: It is assumed that all room surfaces seen by the indoor surface of the glass are at the temperature of the room air. Convection heat transfer between the air and glass surfaces is assumed to follow the equation:

$$h = C\Delta t^{0.25} \qquad \text{(A-1)}$$

where

h = surface conductance for natural convection, Btu per (hour) (square foot) (Fahrenheit degree).

t = temperature difference between glass surface and room air, Fahrenheit degrees.

C = a constant: 0.27 for vertical surfaces, 0.38 for horizontal with heat flow upward.[5]

Because the emissivity of ordinary room surfaces is of the order of 0.90 to 0.95 for low temperature radiation and because the ratio of the room surface area to glass area is large, the indoors has been treated as a black body. The emissivity of glass in these calculations is taken as 0.938[17]. With these conditions and 0 F outdoors, the surface conductances for combined radiation and convection vary from about 1.55 Btu per (hr) (sq ft) (F deg) with low glass temperatures to about 1.40 at glass temperatures within 10 deg of room temperature. For sunlit glass under summer conditions surface conductances vary from about 1.00 Btu per (hr) (sq ft) (F deg) at sunrise to 1.80 (for glass D facing west, at the time of peak heat gain).

A room temperature of 70 F has been used in calculating the U values, but the resulting coefficients are applicable to any temperature in the comfort zone.

All heat gain data for sunlit glass are based upon an indoor temperature of 75 F.

Outdoor Design Conditions—Overall Coefficients: For the computation of overall coefficients the design outdoor air temperature has been taken as 0 F. This temperature level affects both the indoor surface conductance and the radiation exchange with outdoor surroundings. Higher air temperature leads to higher overall coefficients of heat transfer, so that with a 30 F outdoor temperature the U value for single glass at design convection conditions outdoors is 1.20 as compared to 1.17 Btu per (hr) (sq ft) (F deg) at 0 F, and 0.65 for double glass with quarter-inch air space as compared to 0.61.

Because peak load conditions normally occur at night the influence of solar radiation has been considered nil. However, the basic data are available for computing its effect. Appendix C presents a method of computing heat gain or loss due to the combined effect of temperature difference and absorbed solar radiation.

Consideration has been given to the effect of radiation exchange with outdoor surroundings. For design conditions it has been assumed that the sky is clear. The data for radiation from a clear sky have been based upon the work of Brunt[6], who has

correlated the observations of Dines and Dines made in England over a period of several years. The low temperature radiant energy emitted by the clear sky toward a horizontal surface, R_{ih}, Btu per (hour) (square foot), is given by the following equation:

$$R_{ih} = R_{ib}\,(0.55 + 0.056\sqrt{M}) \quad \text{(A-2)}$$

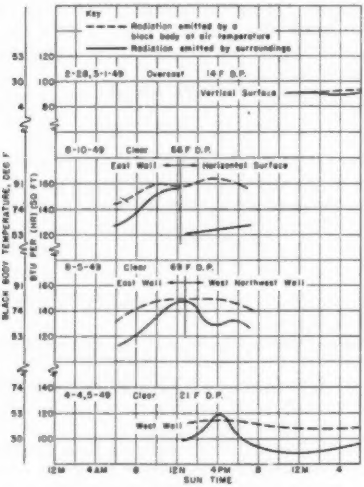

FIG. A-1. LOW TEMPERATURE RADIANT ENERGY EMITTED BY OUTDOOR SURROUNDINGS

It can be shown that the radiation from a clear sky that is received by a vertical surface, R_{iv}, Btu per (hr) (sq ft), is given by the following equation:

$$R_{iv} = R_{ib}\,(0.30 + 0.028\sqrt{M}) \quad \text{(A-3)}$$

where

R_{ib} = black body radiation at the temperature of the air, t Fahrenheit degrees at ground level =

$$0.173\left(\frac{t + 460}{100}\right)^4$$

M = vapor pressure, millibars, at ground level.

Observations made at the A.S.H.V.E. Laboratory with a special radiometer show that, in the absence of solar radiation and with reasonable wind velocities, the ground itself radiates substantially as a black body radiating at air temperature. Since the angle factor of the ground with respect to a vertical plane is 0.5, the radiant energy contribution of ground and sky to vertical surfaces is

$$R_{iv} + 0.5\,R_{ib} \quad \text{(A-4)}$$

If the sky is overcast the entire outdoor surroundings radiate essentially as a black body at air temperature. Clouds are black body radiators and, if the overcast is low, the cloud base is lttle different in temperature from the air temperature at ground level.

Fig. A-1 illustrates typical measurements made at the A.S.H.V.E. Laboratory of the radiant energy emitted by the surroundings to vertical and horizontal building surfaces. For example, data for February 28 and March 1, 1949, illustrate the fact that the outdoors with overcast sky radiate nearly as a black body at air temperature. The values for August 10, 1949 for energy received from a clear sky by a horizontal surface are within 10 percent of the values computed by Equation A-2 while the nightime values shown for April 4, 1949 are in good agreement with Equation A-4.

The design surface conductance for convection only has been rather arbitrarily taken as 4.0 Btu per (hr) (sq ft) (F deg). This value corresponds to the average value for a smooth 10 ft long surface swept by a parallel non-turbulent air stream of 15 mph as determined by wind tunnel tests[11] made at the A.S.H.V.E. Research Laboratory. When combined with radiation, the surface conductance is approximately 6.0, a figure which has been commonly used in heat loss calculations. It is important to note that the surface conductances for convection and radiation *combined* depend upon the thermal resistance of the glass. For example, for single glass it is 5.55 and for double glass with quarter-inch air space it is 6.62 Btu per (hr) (sq ft) (F deg).

Outdoor Design Conditions—Sunlit Glass: An air temperature curve which has a maximum value of 95 F and a minimum of 74 F has been used as a basis for the convection and radiation gain tables for sunlit glass. This curve has been used in the tables of heat gain through walls and roofs published in Chapter 12 of THE 1950 GUIDE[18].

Radiation exchange between vertical surfaces and the outdoor surroundings by day is complicated by the fact that the sun heats the ground and nearby buildings. Numerous measurements, illustrated by Fig. A-1, have been made at the Laboratory and show some correlation with dewpoint. However, these data are insufficient to suggest design values. Therefore, convection and radiation heat transfer *have been combined* in an arbitrary value of 4.0 Btu per (hr) (sq ft) (F deg) for the outdoor surface conductance. As stated in the text, the principal virtue in using this value is that it is consistent with present practice in estimating heat gain through walls and roofs.

Solar Intensity Values: The solar data given in Tables 12 and 13 of the text have been derived from several sources. The values of direct solar radiation for clear atmospheres are the suggested values of Parry Moon[7]. These values are within 5 percent of the values derived from the suggested sol-air design data[8] for Lincoln, Neb. They have been used in preference to the latter because of their acceptance as a standard and because the solar energy distribution is also standardized. The values for an industrial atmosphere were derived from suggested sol-air design temperatures[9] for New York City for a horizontal plane and absorptivity of 1.0. The solar component was found by subtracting from the sol-air temperature the dry bulb temperature which was not exceeded more than 5 percent of the time in a 10-year period in the summer months. This component multiplied by 4 gave the total solar intensity on a horizontal plane. These values were then divided into direct and diffuse components.

The diffuse solar radiation values were derived from an analysis of a considerable number of selected recordings of the daily cycle of total, direct, and diffuse radiation received on a horizontal plane at Cleveland. A family of curves was developed, each of which gave the ratio of direct to diffuse radiation on a horizontal plane as a function of solar altitude. The parameter for this family of curves was the value of the direct normal solar intensity at air mass 2 (solar altitude of 30 deg). Data were then analyzed to develop ratios of diffuse radiation on vertical planes to that on a horizontal plane. All values of diffuse radiation for vertical surfaces include an uncertain amount of ground reflection. Examination will show that they are influenced by the position of the sun.

APPENDIX B

THERMAL PROPERTIES OF GLASS

Transmittance Data—Flat Glass: The transmittance values for glasses A, B, C, and D and their various combinations are given in Table B-1 for use in calculating gain for other exposures and latitudes. The absorptance values are given in Table B-2. These were computed from the spectral transmittance data of four samples studied at the Laboratory for the solar energy distribution given by Moon[7] for an air mass of 2. The same energy distribution was used for all solar altitudes and for both clear and industrial atmospheres. This procedure is not exact but is believed to be a practical method of treating a very complex problem and one that is not subject to great error. As stated in the text, the fact that the absorptance increases as transmittance decreases causes the total gain to change only a little, although the distribution between gain due to transmitted solar energy and that due to convection and radiation from the indoor glass surface is altered.

These transmittance values will be found to agree with experimental data given in A.S.H.V.E. RESEARCH REPORTS [1,2] within the limits of energy distribution effects. Transmittance data for diffuse solar radiation are based upon test results since there is no standard energy distribution for this component of solar radiation. It is known, and test results bear out the fact, that the energy on clear days is concentrated in the visible portion of the solar spectrum. Absorptances for diffuse solar radiation were calculated for the Moon[7] energy distribution for air mass of 2. Each value was then adjusted by the difference between the observed and the computed transmittances for diffuse solar radiation.

Transmittance Data—Glass Block: The transmittance data for the block patterns listed in Table 2 were taken from test data given in an A.S.H.V.E. RESEARCH REPORT[4] presented in 1949. Reference to the spectral transmittance values of samples taken from the block (see discussion of this paper) will show that differences in solar energy distribution will not greatly alter the transmittance values.

Air Space Conductances: The recent correlation by Jakob[19] of the data of Mull and Reiher has been used as the basis for convection heat transfer across air spaces of ½

TABLE B-1—FRACTION OF INCIDENT SOLAR RADIATION TRANSMITTED BY FLAT GLASS

GLASS	SINGLE SHEETS				TWO SHEETS			
	A	B	C	D	A+A	B+B	B+C[a]	C+C
Incident Angle	For Direct Solar Radiation							
0	0.90	0.77	0.41	0.21	0.81	0.60	0.35	0.24
20	0.90	0.77	0.41	0.21	0.81	0.60	0.35	0.24
40	0.89	0.75	0.39	0.20	0.80	0.58	0.33	0.22
50	0.87	0.72	0.37	0.19	0.77	0.55	0.32	0.20
60	0.82	0.67	0.33	0.17	0.71	0.50	0.28	0.18
70	0.70	0.57	0.28	0.13	0.59	0.39	0.21	0.13
80	0.44	0.36	0.18	0.08	0.29	0.18	0.10	0.05
90	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
	For Diffuse Solar Radiation							
	0.82	0.69	0.40	0.22	0.72	0.56	0.32	0.25

[a] Sheet C is outdoor sheet.

TABLE B-2—FRACTION OF INCIDENT SOLAR RADIATION ABSORBED BY FLAT GLASS

	SINGLE SHEETS				TWO SHEETS[a]			
GLASS	A	B	C	D	A+A	B+B	B+C[b]	C+C
Incident Angle	For Direct Solar Radiation							
0	0.02	0.16	0.53	0.74	0.02–0.02	0.17–0.10	0.54–0.04	0.54–0.15
40	0.02	0.17	0.55	0.75	0.02–0.02	0.18–0.11	0.55–0.04	0.56–0.15
60	0.02	0.19	0.56	0.74	0.02–0.02	0.20–0.12	0.58–0.04	0.57–0.13
75	0.02	0.18	0.49	0.62	0.02–0.02	0.21–0.10	0.53–0.03	0.51–0.09
90	0.0	0.0	0.0	0.0	0.0 –0.0	0.0 –0.0	0.0 –0.0	0.0 –0.0
	For Diffuse Solar Radiation							
	0.02	0.16	0.50	0.68	0.02–0.02	0.17–0.09	0.53–0.04	0.51–0.11

a First value for outdoor sheet, second value for indoor sheet.
b Sheet C is outdoor sheet.

in. or greater. The values are substantially lower than those of other investigators, possibly, as Jakob suggests, due to the absence of edge heat transfer. However, when radiation exchange between parallel glass plates is included, the differences between the values used in preparing these tables and those of THE GUIDE are not great.

Thermal Conductivity: The thermal conductivity of glass has been taken as 6.0 Btu per (hr) (sq ft) (F deg per in.). Glass thickness has been taken as ⅛ in. (double strength glass). However, the thermal resistance of the glass is so low that the values may be used for other thicknesses with but little error. In computing the heat gain through sunlit flat glass the thermal resistance of the glass sheet has been neglected.

Conductances of Glass Block Panels: These data have been taken directly from Table 5, page 5 of an A.S.H.V.E. RESEARCH REPORT[4]. These is some evidence that these values vary with surface conductance but there are not enough data from which to draw conclusions regarding the variation.

APPENDIX C

COMPUTATION METHODS

Overall Coefficients: For the steady state, the heat flow through a sheet of glass in the absence of solar radiation is given by the expression:

$$q = h_i\,(t_i - t_{gi}) = k/L\,(t_{gi} - t_{go}) = h_o\,(t_{go} - t_o) + R_x \quad . \; . \; . \; . \quad \text{(C-1)}$$

where

q = rate of heat flow, Btu per (hour) (square foot).

h_i = the indoor surface conductance for radiation and convection combined, Btu per (hour) (square foot) (Fahrenheit degree).

h_o = the outdoor surface conductance for convection, Btu per (hour) (square foot) (Fahrenheit degree).

k = thermal conductivity of glass, Btu per (hour) (square foot) (Fahrenheit degree per foot).

L = glass thickness, feet.

t_i = indoor temperature, Fahrenheit degrees.

t_o = outdoor air temperature, Fahrenheit degrees.

t_{gi} = temperature of indoor surface of glass, Fahrenheit degrees.

t_{go} = temperature of outdoor surface of glass, Fahrenheit degrees.

R_x = net radiant energy exchange between glass and outdoor surroundings, Btu per (hour) (square foot).

For glass block walls and double glass, $C(t_{gi}-t_{go})$ was used in place of $k/L\ (t_{gi}-t_{go})$, where C is the conductance of the section, Btu per (hour) (square foot) (Fahrenheit degree). Because of the dependence of h_i and R_x on t_{gi} and t_{go}, Equation C-1 requires a trial and error solution. In this instance a curve plotting method was used.

Convection and Radiation Gain from Sunlit Flat Glass: The computation of convection and radiation gain from sunlit glass is more complex so that it was necessary

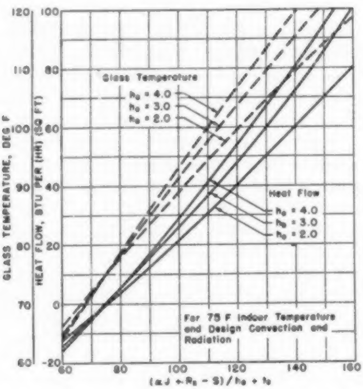

Fig. C-1. Convection and Radiation Heat Gain and Temperature for Single Glass

to simplify the heat balance equations by ignoring the small influence of the thermal resistance of the glass sheet. Equation C-1, now including terms for solar energy absorption and heat storage, becomes:

For one sheet:

$$q = h_i\,(t_{g1} - t_i) = \alpha_1\,J + R_x - S_1 - h_o\,(t_{g1} - t_o) \quad \text{. . . . .} \quad \text{(C-2)}$$

For two air-spaced sheets:

$$q = h_i\,(t_{g1} - t_i) = \alpha_1\,J - h_s\,(t_{g1} - t_{g2}) - S_1 \quad \text{. . . . . .} \quad \text{(C-3)}$$

where

t_{g1} = temperature of a single sheet (or of the indoor of two sheets), Fahrenheit degrees.

t_{g2} = temperature of the outdoor glass sheet, Fahrenheit degrees.

α_1 = fraction of the incident radiation absorbed by a single sheet (or by the indoor of two sheets).

J = incident solar radiation, Btu per (hour) (square foot).

S_1 = rate of heat storage of a single sheet (or of the indoor of two sheets) Btu per (hour) (square foot).

h_s = air space conductance, Btu per (hour) (square foot) (Fahrenheit degree).

These heat balances must also be solved by trial and error. For a fixed indoor temperature, however, curve families with h_o as the parameter can be drawn with heat flow and t_{g1} as functions of the expression $\dfrac{\alpha_1 J + R_x - S_1}{h_o}$. Since R_x and h_i are dependent upon t_{g1} and t_{g2}, a trial and error solution is required. Fig. C-1 shows curves for a single sheet of glass for an indoor temperature of 75 F, the design indoor convection equation and several values of h_o. Fig. C-2 presents a set of curves for two glass sheets for the same conditions with h_o fixed at 4.0 Btu per (hr) (sq ft) (F deg).

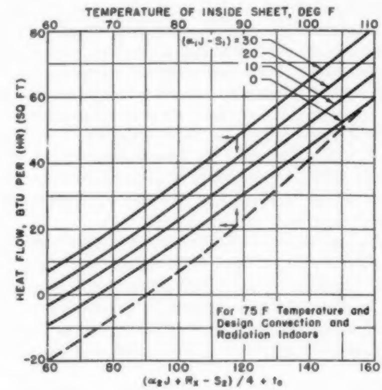

FIG. C-2. CONVECTION AND RADIATION GAIN AND TEMPERATURE OF INDOOR SHEET FOR DOUBLE GLASS

The curve parameter is the amount of solar radiation absorbed by the indoor sheet of glass. The subscript 2 on α and S refers to the outdoor sheet. A net work chart has been constructed to express Equations C-1 and C-2 in general terms.

In setting up Tables 5 and 7, no attempt has been made to treat R_x because of the inadequacies of the present data. Instead a surface conductance of 4.0 Btu per (hr) (sq ft) (F deg) has been used to approximate the combined results of convection and radiation. The rate of heat storage, S_1, has been taken into account by adding algebraically to the computed heat gain the rate of heat storage as determined experimentally. This quantity is insignificant for glass A but of appreciable magnitude for the heat absorbing glasses.

Convection and Radiation Gain from Sunlit Glass Block: Of several attempts to devise a rational method of determining periodic heat flow through glass block walls by use of either basic thermal properties or test data, or both, only one was reasonably successful. This method, used in computing the values in Tables 9 and 11, required a series of heat balances, in which the radiation absorbed by the block in a one-hour period was balanced against the stored energy and the heat lost at the indoor and outdoor surfaces in the same period. A trial and error solution gave the temperature of the indoor surface of the glass at the end of the period. This temperature was then

the starting point for the next period. Necessary to the computations was a relationship between the indoor and outdoor air temperature difference, and the indoor and outdoor glass block surface temperature difference. Empirical curves which gave this relationship as a function of sun time were developed for each orientation from fixed position tests made on the various panels. Absorption coefficients for the blocks were computed from test data by means of heat balances. It was found that the six patterns could be placed in three groups as shown in the design tables.

Although the separate influence of such variables as wind, solar intensity and radiation exchange could not be entirely accounted for in fairing the empirical temperature difference curves, nevertheless the computed design data showed satisfactory correlation with experimental measurements in those tests in which design conditions were closely approximated. Computation was simplified when it was found that the heat flow from the indoor surface for a fixed indoor temperature and heat capacity was a linear function of a three-term sum which involved the absorbed solar radiation, the surface temperature difference, and the outdoor air temperature and surface conductance.

In concluding this discussion of computation methods, some comments on the effect of ignoring radiation exchange with the outdoor surroundings are appropriate. In general, it may be said that, whenever summer temperatures prevail and a wall is not irradiated by the direct sun, its outdoor surface temperature usually indicates that it is gaining heat by convection from the air. However, it will often be found that it is losing heat at a considerably greater rate by radiation to the sky and ground. Hence the design values of heat gain by convection and radiation for flat glass are somewhat higher for those periods when they are not directly exposed to the sun than would be the case if radiation exchange had been included.

On the other hand the temperature difference curves for the glass block are to some degree affected by radiation exchange. Therefore, the values for convection and radiation gain from flat glass and glass block are not strictly comparable. At the time of peak heat gain the loss by radiation is less in proportion to that by convection, so that the use of a design surface conductance of 4.0 Btu per (hr) (sq ft) (F deg) approximates fairly well the combined effect of a low wind velocity and radiation exchange.

DISCUSSION

J. N. LIVERMORE, Detroit (WRITTEN): This paper is a highly valuable contribution to precision in calculating heat gain through glass. It has been carefully written and reflects sound research procedure throughout.

As you may know, a joint committee, composed of members from the TAC on Heat Flow through Glass and the TAC on Cooling Load, has been assigned the task of recommending ways to present the information in this paper in a form readily usable by the practicing engineer. As chairman of this joint committee I would like to take this opportunity to invite your suggestions in this matter.

Whatever form these new data take, I would like to remind you also that a great job of condensing, perhaps with the use of empirical data, will have to be done before this new information can appear on the pages of THE GUIDE. It is very likely too, that for any but the most ordinary glass and solar intensity combinations, reference must still be made to the original paper.

For this reason I would like to suggest that the authors present a supplement to the paper in which they show practical examples in which calculations demonstrate the proper application of the data they have presented.

R. W. MCKINLEY, Pittsburgh (WRITTEN): I am convinced that this research has produced a great deal more helpful information than the authors, in their modesty, have claimed. My purpose is to ask them to set forth this helpful information in simple language so that the application engineer, architect and designer may more quickly benefit from their studies.

Granting that the authors have specialized in laboratory research rather than application engineering, I wish, nevertheless, that they had started their paper with a simple tabulation of suggested design standards and procedures based on the knowledge they have gained through their work at the laboratory. Since the designer must choose a U value, estimate heating load, cooling load, lay out air distribution, choose ducts, fans, controls, and a multitude of related items, I am sure he would appreciate any application recommendation the author might make. For example:

1. What overall coefficients are preferable and under what circumstances?
2. Table 3 gives certain data for flat glass. Would it not be possible to add directly comparable data for glass block so that the designer would be better able to compare the performance of the two materials?
3. In the section Discussion of New Data, the authors discuss some of the reasons for differences between their U values and those determined by others. If this is a matter of precision of measurement unrelated to field conditions, I would like the authors to suggest under what design conditions they would use each set of values. Such things as these certainly would not take anything away from the scientific validity of the research and would be really helpful to those who are called on to make design decisions with limited opportunities to dig deeply into the research.

D. D'EUSTACHIO, Port Allegany, Pa. (WRITTEN): The precise measurement of heat flow and thermal conductivities is generally considered a difficult task. Much of the difficulty arises from unsuspected heat leaks and lack of true equilibrium conditions. As the authors' earlier papers show they have taken precautions against these types of errors.

In practical situations another problem and possible source of error arises. This has to do with defining the environmental conditions which affect the heat flow. Here again the authors have been careful to describe their apparatus and state exactly how it was operated. It is, perhaps, unreasonable to expect that a paper of this character should also take the next step, namely, of setting up a procedure to enable the engineer and architect to transpose the data to the room or building with which he has to deal. This essentially requires the making of proper corrections and allowances for an environment that differs from that of the experimental set-up. The importance of doing this properly and the inherent difficulty in the process is well illustrated by the comparison the authors make between their overall U values for glass block panels and those in THE GUIDE 1950. They state that the 10 to 20 percent difference in these two sets of values does not arise from the differences in panel conductances (here there is excellent agreement) but from the differences in evaluation of the characteristics of the indoor and outdoor environment.

Because this problem is very important and because its solution must be based on the kind of knowledge that comes from much field experience, perhaps the TAC on Heat Flow Through Glass can obtain the services of an experienced practicing engineer to suggest a procedure and to guide the work of arranging the new data in a manner that will best serve users of THE GUIDE.

E. W. CONOVER, Detroit (WRITTEN): It is evident that the authors have spent considerable time in evolving theories and computing the mass of data given in this paper. After doing so, they step out of character and beyond the function of the committee under which they worked by devising Section C of Table 1. This table is not only incorrect but would seem to have no place in this paper.

Normally, the heating engineer uses the masonry opening for computing heat transmission because he does not know what kind of window is going to fill the opening. Also he is not interested in determining the percentage of exposed glass but wants a single factor or coefficient that will average his conditions. A small plus or minus differential in computed heat loss through windows is of little importance since glass transmission usually represents only 20 to 30 percent of total heat requirements. The coefficient U, being based on a design condition seldom encountered, and then only at

night when there is no sky or solar radiation, appears to have a considerable factor of safety in it.

In the discussion of the data it is stated, after referring to Fig. 1, that the indoor surface conductances for the calorimeter are of the order of 1.20 as compared to 1.40 to 1.50 Btu for design conditions. I cannot reconcile these figures with the curves of Fig. 1.

For the lower curves of Fig. 2 a single test result is shown for each curve, one being at a considerable variance. It seems unlikely that a single test point substantiates the shape and position of a computed curve.

Table 8 shows the heat gain for glass blocks I and II in south elevation to be lower at noon than at 11:00 a.m. or 1:00 p.m. while other blocks show higher. It appears that this is in error. In this paper, as well as others by the same authors, we have looked in vain for test readings of glass temperatures, particularly those tests conducted on sunlit glass during cold weather. It seems reasonable that, due to heat storage, the transmission losses during winter daylight hours would be considerably less than for design conditions and might even be zero.

The approach to solar energy transmittance of the various types of glass seems to have been made solely from the standpoint of its effect on the summer cooling load. A similar, but opposite effect is evident on the winter heating load. I question whether the saving in summer cooling by reducing the admittance of solar energy is not counterbalanced by the increased winter heating load, especially in northern climates where the heating season is longer than the cooling season. This would seem to argue in favor of clear sheet glass with some type of movable control to keep out the solar energy only when desired rather than a type of glazing that reduces solar radiation at all times.

W. C. RANDALL, Detroit: I note in Table 1 that the outdoor conditions are specified at zero degrees Fahrenheit, clear skies, no solar radiation and a convection conductance of 4 Btu per (sq ft) (F deg temperature difference). Could the author set up a table for the popular or average condition? Why put down only the optimum condition.

I assume that the 4 Btu (per hr) (sq ft) is somewhat synonymous to a 15-mile wind, and if we could set up a condition for still air with no wind, certainly that is the condition that prevails on at least half of the building and something less than 15 mph prevails a great deal of the time.

Several years ago Professor Emswiler and myself presented a paper on Room Surface Temperatures of Glass in Windows, in which we indicated a figure of 0.86 for still air conditions as compared with the 1.13 normally accepted for 15-mile wind conditions. In other words, so we do not over-design, it seems to me another table might be set up.

On Section C, Application Factors for Windows, we have for wood sash 80 percent glass, 90 percent of the factor, and for 60 percent glass 80 percent of the factor. You can then solve for the wood in that case. It checks out about 0.65 in both cases. If you set that up for the wood sash in the next table, for double glass 80 percent, the 0.95 goes to 1, and for the 60 percent glass that factor of 0.85 goes to 1.

I would interpret under single glass that comparing metal sash with 80 percent glass and wood sash with 60 percent glass for any given opening, we have 33⅓ percent more glass in the metal window with but 25 percent more heat loss. In the column under double glass, if we change the factor of 0.85 to 1, as suggested, a similar conclusion can be drawn, that is 33⅓ percent more glass in the metal window for 25 percent more heat loss compared with the wood windows. Incidentally, the figures on windows with storm sash indicate but 25 percent more heat loss through the metal sash opening with 33⅓ percent more glass than in the wood sash using the 60 percent glass figure.

I think the use of the tables would be more apparent if there could be set up, for the glass storm sash columns, a factor that was correlated with the single glass coefficient 1.17.

LINN HELANDER, Manhattan, Kans.: What was the relative humidity of the atmosphere under the conditions for which the radiation losses from the unshaded glass were evaluated, and what effect on these radiation losses would variations in the humidity of the atmosphere have?

AUTHORS' CLOSURE: Both Mr. McKinley and Mr. Randall have raised questions concerning the *U* values given in Table 1. In the opinion of the authors, the particular values given seem to represent a reasonable combination of weather and indoor conditions on which to base heating system designs. Radiation losses are greatest with clear skies, at which time the wind velocity is not likely to be excessive. If the designer has available exact data on weather and surface conductances for his particular problem, *U* values for most of these conditions can be found in Figs. 1 and 2.

Dr. D'Eustachio and Mr. McKinley point out differences in *U* values for glass block as given in this paper and those appearing in the literature. Investigation will show that the panel conductances as determined by this research are little different from those obtained by others. The *U* values, on the other hand, depend upon the design heat transfer conditions at the indoor and outdoor surfaces. The design conditions which we have used are quite different from conditions in some hot box tests where the box surfaces were reflective. This results in low surface conductances for radiation and consequently low overall coefficients as compared with the design data used in this paper.

Messrs. Randall and Conover point out the need for data on solar heat gain in winter and state that certain types of glass used to reduce cooling load in summer reduce the gain that is desirable in winter. The data given in the paper are of use in estimating these gains, provided the solar intensity and angle data are known. If the solar gain from glass in winter is a factor of importance, the engineer or architect should make a careful economic study of various methods of glazing light admitting areas, balancing the initial and maintaining cost of the glazing and shade combination against intial and seasonal operating costs of the heating and cooling equipment. The factor of occupant comfort should not be ignored in such an analysis.

The application factors of Section C, Table 1 on overall coefficients for windows are criticized by Messrs. Randall and Conover. The paper states that these are approximate values, and although no great precision is attached to the values, they are believed to be of the correct order of magnitude. The values have been provided in order to answer the inevitable question, *How can the U values for glass sheets be applied to actual windows?* The design engineer is at liberty to use or ignore the application factors or to select his own *average* value.

The application factors were determined by comparing *U* values for glass sheets with *U* values for windows, all tests having been made in the same hot box under identical conditions. For example, in one group of tests summarized in reference 10, the *U* values given for a single glass sheet, single glazed wood casement, single glazed double hung, and single glazed metal casement windows were 0.76, 0.58, 0.56 and 0.76, respectively, for glass areas of 100, 58, 54 and 76 percent, respectively. By interpolation, the factor for 80 percent wood sash is 0.885, or roughly 0.90 as given in the paper. Values from another set of tests discussed in reference 10 gave a ratio of 1.14 for wood and metal sash windows of the same light size. This compares favorably with the ratio 1.11 of the factors for wood and metal sash windows of 80 percent glass area. Other factors were derived in similar fashion. Factors computed from different sets of test data compared favorably.

All curves of Fig. 1 and Fig. 2 are computed curves based on conditions described in the text and were not established by the few test points shown as Mr. Conover

suggests. The curves denoted by the phrase *calorimeter convection* are an attempt to show why U values, as determined by test and reported in reference 3, are not the same as those suggested for design use. The calorimeter convection data on which these calculations were based were given in reference 3. This reference gives glass temperatures in winter, during day tests and during night tests. Most of these tests were performed with glass A. One made during the afternoon with the glass kept out of the sun gave an average U value of 0.87, another 0.94 (points for 1-23-48 and 1-26-48 in Fig. 1). The respective outward heat flows of about 50 and 35 Btu per (hr) (sq ft) due to temperature difference were partly balanced by gains due to transmitted diffuse solar radiation of about 25 Btu per (hr) (sq ft) near noon and of about 10 Btu at 4:00 p.m.

The Table 3 heat gain values for glass blocks I and II are not in error. The transmittance of these blocks is higher at 11:00 a.m. (or 1:00 p.m.) than at noon because of the effect of azimuth and altitude on transmittance. The reverse is true for all the other block patterns at these hours. Inspection of the transmittance data for glass block presented in a paper given before the Society in June 1949 will bring out this point.

With regard to Professor Helander's question as to the effect of relative humidity on radiation losses from glass, we have used a value of 4.0 Btu per (hr) (sq ft) for combined outdoor convection and radiation in preparing the heat gain tables. We have obtained some data on radiation from the atmosphere as illustrated in Fig. A-1. In days of high dew point we find the radiation from the atmosphere to be greater than on days of low dew point. Everything else being equal this would reduce the heat lost in the radiation exchange between the glass and its surroundings, and thereby increase heat *gain* by an air conditioned space. This is a complex problem that cannot be fully answered in this discussion. We are in the process of preparing a paper on this phase of heat transfer.

Mr. Livermore asks that a supplement to the paper be prepared to show practical examples demonstrating the application of the data presented. It is planned to prepare a bulletin summarizing the glass research in which material of this type can be included as well as certain other data that could not be included in the paper but which are necessary in handling some problems.

1400

No. 1400

EFFECTIVE SOLAR ABSORPTION OF VARIOUS COLORED PAINTS

By R. H. Heilman* and R. W. Ortmiller**, Pittsburgh, Pa.

At THE suggestion of the A.S.H.V.E. Technical Advisory Committee on Cooling Load, solar absorption tests were run to determine the absorption coefficients of numerous paints exposed to the direct radiation of the sun.

There is considerable information available in the literature on absorption coefficients or emissivities of numerous surfaces for radiation at various wavelengths; also there is some information giving relative heat absorptions and temperatures attained by surfaces exposed to the sun. There is very little information, however, giving actual coefficients for surfaces exposed to the sun. The tests reported in this paper were run primarily to obtain these coefficients and to make a comparison between the absorption coefficients of paints obtained by direct exposure to the sun and those obtained by exposure to radiation at wavelengths approximating the wavelength corresponding to the maximum intensity of the sun.

Investigators have found that some surfaces exhibit selective reflection characteristics, the reflection coefficients varying widely for comparatively small differences in wavelength of the incident radiation. For instance, Coblentz[3] obtained reflection coefficients for chromium oxide of 24.1, 27.0 and 44.6 at wavelengths of 0.54, 0.60 and 0.95 microns respectively.

Description of Apparatus and Paints

The apparatus used in making the tests consisted of a sensitive thermopile containing 96 very thin thermo-elements connected in series and previously described by Heilman[1], a portable precision potentiometer, and a rigid mounting device which held the samples and the thermopile in a definite position relative to each other. The assembly of the sample holder, the thermopile and a sun sight was mounted on a rotating member so that the face of the sample could be held perpendicular to the sun's radiation at all times during a measurement. A photograph of the test assembly less the potentiometer is shown in Fig. 1. An insulated shield was placed around the assembly to protect the sample and thermopile from air currents.

The paint samples (most of which were supplied by the A.S.H.V.E. Research Laboratory) were applied to aluminum foil 0.00035 in. thick in order to keep the

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[1] Exponent numerals refer to References.

Presented at the Semi-Annual Meeting of The American Society of Heating and Ventilating Engineers, Muskoka, Ontario, Canada, June 1950.

mass of the receiving surface at a minimum. The underside (the side of the aluminum foil exposed to the thermopile) was painted with a flat black lacquer. The emissivity of the black lacquer as measured at atmospheric temperatures was 0.95. A standard receiving surface of aluminum foil having flat black

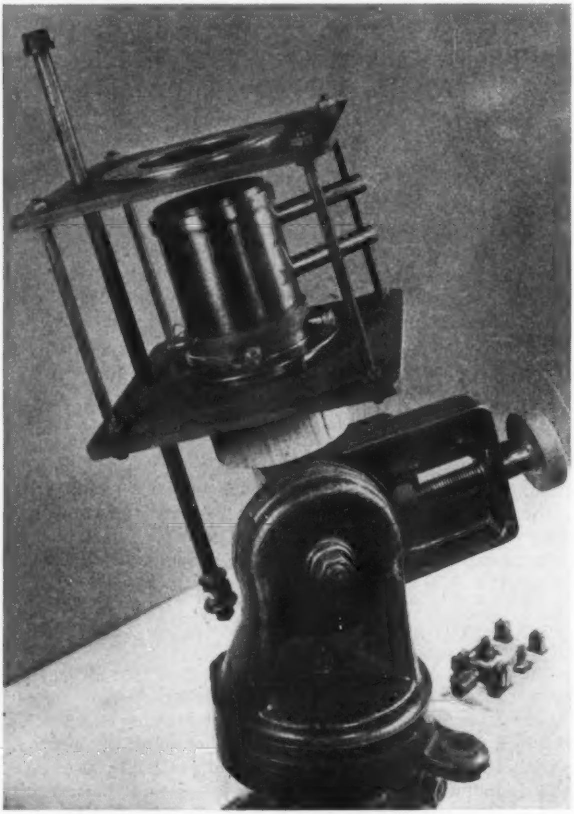

FIG. 1. SOLAR ABSORPTION APPARATUS

lacquer on both sides was used alternately with the test samples to minimize possible variations in the sun's intensity due to invisible clouds or moisture and to change in altitude of the sun. The time required to make the observations on a given sample was usually less than 5 min. The samples of the regular or white asbestos paper and black asphalt-saturated paper were approximately 0.025 in. thick. Their *black heat* emissivities were the same, 0.93. The analyses of most of the paints tested are shown in Table 1.

TABLE 1—ANALYSIS OF PAINTS TESTED

	(1) SLATE	(2) YELLOW	(3) GREEN	(4) BROWN	(5) WHITE ENAMEL	(6) WHITE	(7) WHITE
Pigment by weight	64	63	38	44	38	61.5	59.5
Vehicle by weight	36	37	62	56	62	38.5	40.5
Pigment by weight:							
Leaded Zinc Oxide	49	44	—	12	—	—	—
Basic Sulphate White Lead	19	16	—	4	—	—	—
Zinc Oxide	30	28	6	8	—	29	—
Titanium Magnesium	40	43	—	—	—	62.1	—
Titanium Calcium	—	—	—	—	—	—	57.3
Magnesium Silicate	5	5	54	48	—	—	—
Basic Carbonate White Lead	6	8	—	—	—	—	—
Chrome Green CP	—	—	40	—	—	—	—
Ferric Oxide	—	—	—	40	—	—	—
Titanium Pigment	—	—	—	—	100	—	—
Calcium Carbonate	—	—	—	—	—	8.9	—
Lithopone	—	—	—	—	—	—	39.5
Silicates	—	—	—	—	—	—	3.2
Vehicle by weight:							
Linseed Oil	70	69	33	30	—	69.4	—
Drier	15	15	—	—	—	1.3	27.
Mineral Spirits	15	16	—	—	—	—	65.9
Varnish[a]	—	—	67	70	—	—	—
Spar Varnish	—	—	—	—	100	—	—
Petroleum Thinner	—	—	—	—	—	29.3	—
Resin	—	—	—	—	—	—	7.1

[a] Resin 8, Vegetable oils 40, Mineral spirits 52.
Other paints tested: Red roof paint (8) Aluminum paint (9) and Aluminum paint with asphalt vehicle (10).

CALCULATIONS OF TEST RESULTS

The thermopile had previously been calibrated so that the emissivity of a surface could be determined from measurements of the electromotive force (emf) generated by the thermopile, the emf generated by a thermocouple attached to one of the cold junctions, the temperature of the surface whose emissivity was being measured, and the temperature of the receiving surface of the thermopile. The temperature of the receiving surface could be calculated from the temperature of the cold junction and the number of hot junctions connected in series. The constant of the thermopile was 0.118×10^{-10}.
Then

$$e = \frac{D}{0.118 \times 10^{-10} (T_1^4 - T_2^4)} \qquad (1)$$

where

e = emissivity of surface.
D = net electromotive force generated.
T_1 = temperature of surface, Fahrenheit, absolute.
T_2 = temperature of receiving surface, Fahrenheit, absolute.

Also the heat loss by radiation, q_r, Btu per (hr) (sq ft) between the two surfaces is:

$$q_r = 17.4 \times 10^{-10}\, e\, (T_1^4 - T_2^4) \quad \ldots \quad (2)$$

substituting for value of e (from Equation 1),

$$q_r = 17.4 \times 10^{-10} \frac{D}{0.118 \times 10^{-10}} \frac{(T_1^4 - T_2^4)}{(T_1^4 - T_2^4)} = 17.4 \frac{D}{0.118} = 147.5\, D \quad \ldots \quad (3)$$

In Equation 2, e is the effective emissivity between the two surfaces and is accounted for in the constant of the thermopile. Since the same black surface was exposed to the thermopile at all times, it was assumed that it would measure accurately at all times the radiation loss from the underside of any of the test specimens. The heat loss from the underside of the surface by conduction can be considered to be negligible. The heat loss by convection from the underside would also be very low because of the ¼ in. deep depression in the hardboard plate holding the test sample and the length of travel the air would have to flow over the hardboard plate before it would reach the insulated sides of the assembly.

Temperatures of the underside of the test sample and the receiver of the thermopile were calculated for the white painted sample and the black painted sample. Assuming that the convection loss would be the same as that of a horizontal plate warmer than air facing downward (the actual convection loss would be very much less than this), it was found that the ratio of the total loss from the underside of the test samples to the ratio of the loss by radiation was 2.11 to 2.09, or less than 1 percent. It can, therefore, be assumed that the ratio of the radiation losses is very close to the ratio of the total losses from the underside. Since the object is to determine the effective solar absorption coefficient which is actually equivalent to the total heat transmitted to the underside of the sample, it is not necessary to be concerned with convection and reradiation losses from the top of the sample.

It was assumed that the solar absorption coefficient of the standard black surface was 0.95, the same as the measured *black heat* emissivity. The heat coming through the sample would, therefore, be increased by the ratio 1/0.95 or 1.0525 if the top black surface had been a perfect black body.

The effective solar absorption coefficient, α, then, of a given surface, x, is

$$\alpha = \frac{q_r \text{ (surface } x)}{1.0525\, q_r \text{ (black surface)}}$$

For example, the effective solar absorption coefficient of a white paint as observed is

$$\alpha = \frac{147.5 \times 0.279}{1.0525 \times 147.5 \times 0.63} = 0.42$$

If the assumed solar absorption coefficient for the black surface is high or low, the calculated absorption coefficient for the various surfaces tested will be correspondingly high or low. There was apparently little or no reradiation from the sky, as in most cases a negative reading was obtained when the sample was pointed to the sky away from the sun.

TABLE 2—EFFECTIVE SOLAR ABSORPTION COEFFICIENTS OF VARIOUS PAINTS AND MATERIALS

MATERIAL	CO-EFFICIENT	MATERIAL	CO-EFFICIENT
(1) Slate	0.69	(7) White	0.36
(2) Yellow	0.59	(8) Red Roof	0.86
(3) Green	0.93	(9) Aluminum Bronze	0.39
(4) Brown	0.83	(10) Aluminum Bronze	0.37
(5) White Enamel	0.42	(11) Aluminum Foil Bright	0.41
(6) White	0.51	(12) Regular Asbestos Paper	0.50
(6a) Same after 2 mos. exposure	0.68	(13) Black Asphalt Saturated Paper	0.93

The average of several observations for each material tested is given in Table 2.

The emissivities of some surfaces obtained by other investigators for radiation at a wavelength of 0.6 μ are shown in Table 3.

DISCUSSIONS AND CONCLUSIONS

Both paint samples 5 and 6 had a distinctive yellowish cast. Sample 7 was very white. Sample 6 was tested before and after two months exposure to the weather on the *Mellon Institute* roof. The tests indicate that the solar absorption coefficient can change very rapidly in Pittsburgh's atmosphere. The coefficient obtained on the bright aluminum foils confirms the general belief that aluminum foil on a roof is usually no more effective in reducing the heat load on the roof than a good white paint. This is due to the fact that the aluminum foil is a poor emitter of radiant energy in the infrared region while white paint is a good emitter in this region. The aluminum foil can reradiate to the sky and surroundings only a small portion of the energy it receives from the sun, consequently its surface temperature rises. The white paint can reradiate most of the energy it receives to the surroundings with a tendency for the surface temperature to drop. The net result is very little difference in the surface temperatures of the two materials. Actually the true solar absorption coefficient of aluminum foil is probably very much lower than that of white paint, while practically as far as heat gain is concerned there is very little difference as long as their surfaces remain in their original condition.

A direct comparison cannot be made between the white paints tested in this investigation and the white paints as tested by Coblentz[3] because the composition of the pigments and the vehicles are vastly different. It is reasonable to assume that the paints tested by Coblentz should have lower absorption coefficients as they contained nothing but the pure pigment and linseed oil.

The greatest sources of error in these tests are probably the changing intensity of the solar radiation and the assumption that the solar absorption coefficient is 0.95 for the black standard. The value assumed for this is probably accurate to within ± 3 percent. The rapidity with which the readings were taken should minimize the errors due to changing intensity. However, it is believed that more accurate results could be obtained if the tests were conducted in a region where the intensity is more constant.

TABLE 3—ABSORPTIVITIES OF VARIOUS SUBSTANCES FOR RADIATION OF A WAVELENGTH OF 0.60 MICRONS

Material	Absorptivity	Authority
Bricks		
Light Buff	0.52	Coblentz[3]
Darker Buff	0.60	Coblentz[3]
Red	0.70	Coblentz[3]
Darker & Glazed	0.77	Coblentz[3]
Wire Cut red	0.61	Beckett[4]
Sand lime red	0.72	Beckett[4]
Mottled purple	0.79	Beckett[4]
Clay Tiles		
Machine Made red	0.68–0.71	Beckett[4]
Machine Made dark purple	0.82	Beckett[4]
Concrete Tiles		
Uncolored	0.65	Beckett[4]
Brown	0.85–0.87	Beckett[4]
Black	0.91	Beckett[4]
Metals		
Pure polished iron	0.45	Coblentz[3]
Magnesium	0.30	Coblentz[3]
Zinc, pure polished	0.46	Coblentz[3]
Duralumin	0.53	Coblentz[3]
Monel metal	0.43	Coblentz[3]
Molybdenum	0.55	Hulbert[5]
Nickel, electrolytic	0.40	Hagen & Rubens[6]
Speculum metal	0.39	Hagen & Rubens[6]
Slates		
Blue grey	0.87	Beckett[4]
Dark grey	0.90	Beckett[4]
Miscellaneous		
Galv. iron: white-washed	0.21	Beckett[4]
Asbestos cement: white	0.60	Beckett[4]
Asbestos cement: red	0.66	Beckett[4]
Roofing lead: old	0.81	Beckett[4]
Galv. iron: new	0.66	Beckett[4]
Galv. iron: very dirty	0.89	Beckett[4]
Enameled steel: white	0.47	Beckett[4]
Enameled steel: blue	0.83	Beckett[4]
Enameled steel: red	0.83	Beckett[4]
Enameled steel: green	0.88	Beckett[4]
Weathered asphalt	0.89	Beckett[4]
Bituminous felt	0.89	Beckett[4]
Asphalt pavement (dust free)	0.93	Coblentz[3]
Glazed porcelain	0.28	Coblentz[3]
White marble	0.47	Coblentz[3]
Pigments		
Blue (CO_2O_3)	0.97	Coblentz[3]
Red (Fe_2O_3)	0.74	Coblentz[3]
Green (Cu_2O_3)	0.73	Coblentz[3]
Yellow (PbO)	0.48	Coblentz[3]
White (Al_2O_3)	0.16	Coblentz[3]
White (Y_2O_3)	0.26	Coblentz[3]
White (ZnO)	0.18	Coblentz[3]
White (ZrO_2)	0.14	Coblentz[3]
White Paints Ground in Oil		
White lead	0.25	Coblentz[3]
Zinc lead	0.30	Coblentz[3]
Zinc oxide	0.32	Coblentz[3]

The actual temperature of the sun has been variously estimated at from 10,000-11,500 F. The wavelength at which the intensity is a maximum for these temperatures is approximately 0.5 and 0.43 μ respectively, with most of the energy lying in the wavelength range of 0.1 to 5.0 μ. Since it is known that some substances exhibit selective radiation characteristics and since at least half of the energy emitted from the sun lies in the visible range, it would be desirable to check several identical samples by both methods using, if possible, wavelengths of 0.1 to 5.0 μ in the selective radiation tests.

ACKNOWLEDGMENT

The authors are indebted to Prof. R. C. Jordan of the University of Minnesota for the Bibliography on solar absorption determinations.

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Figuring Solar Heat Gains of Buildings, by William Goodman (tests to determine effect of solar radiation and thermal capacity on heat transmission) (*Heating, Piping & Air Conditioning,* Vol. 10, No. 10, October 1938, pp. 657-659).

A.S.H.V.E. RESEARCH REPORT No. 853—Absorption of Solar Radiation in its Relation to the Temperature, Color, Angle and Other Characteristics of the Absorbing Surface, by F. C. Houghten and Carl Gutberlet (relative heat absorption by different surfaces relative to black oil cloth) (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, pp. 137-152).

Surface Absorption of Heat from Solar Radiation, by F. G. Hechler and E. R. Queer (solar absorption characteristics of various painted surfaces and building materials) (*Refrigerating Engineering,* Vol. 25, No. 2, February 1933, pp. 86-91, 104, 115).

A Study of Temperatures under Sheet Steel Roofs, by H. Giese, W. D. Scoates, and N. C. Ives (experimental determination of temperatures under sheet steel roofs exposed to sun) (*Agricultural Engineering,* Vol. 20, No. 7, July 1939, pp. 267-269).

DISCUSSION

J. P. STEWART, Syracuse, N. Y. (WRITTEN): This paper is a worthwhile contribution to our knowledge of heat flow for cooling load calculations. I suggest that this excellent work be extended to include building materials such as dark, red brick, bright red brick, cream colored brick, natural color concrete, white marble and other common unpainted building materials. Other kinds of paint such as special highly reflective white paint used for signs and road marking may also be tested. Much used colors on Venetian blinds, such as cream or egg shell and light grey, would yield useful data for tests of Venetian blinds.

C. B. BRADLEY, Manville, N. J. (WRITTEN): The paper presents data of interest, reported with care, including a careful description of test conditions, permitting the reader to evaluate the results.

An interesting extension may be made to the authors' comments on the comparison between an aluminum and a white roof. They point out that, due to the low emissivity of aluminum and the high emissivity of white paint in the infra-red region, the aluminum roof is *usually no more effective in reducing the heat load on the roof than a good white paint*. This effect is much more pronounced at night when the roof is radiating to the sky with a very low effective temperature. The white roof, with an emissivity in the neighborhood of 0.9, radiates its heat energy very efficiently and cools off rapidly while the aluminum with a very low emissivity cools off much more slowly. Thus, much of the heat stored in the roof escapes to the surroundings in the case of the white roof, and much less is transmitted into the building.

F. C. HOOPER, Toronto, Can. (WRITTEN): There can be no dispute of the findings of these authors and of other investigators who report effective solar absorption coefficients for bright aluminum surfaces exceeding those for white surfaces. However, I believe the explanation of this phenomenon which is offered to be in error.

Consider a surface normal to the sun's rays in Pennsylvania, where the solar radiation intensity would be approximately 350 Btu per (hr) (sq ft). Suppose this surface reaches an equilibrium temperature of 100 F in 80 F air while seeing a blue sky at 60 F. Even if the surface has a low temperature emissivity of unity, it will reradiate only

$$0.173 \times 10^{-8} [(100 + 460)^4 - (60 + 460)^4] = 45 \text{ Btu per (hr) (sq ft)}.$$

Thus, on the basis of a maximum reradiation the absorptivity will have eight-fold more influence on the net heat gain than has the low temperature emissivity. This suggests that the true absorptivity is controlling.

Moreover, even for a black surface, about as much heat will be lost by convection as by radiation, and reradiation is not the controlling factor in the heat loss to the surface.

Because of this condition, it would appear that substantial differences in apparent solar absorption must be due to differences in the true solar absorptivities of the surfaces. The results of Sieber* and others, obtained by integrating selective radiation tests over the solar energy distribution curve, agree with this interpretation. Curve 1(a), shows Sieber's results for aluminum and for white fire clay, and indicates the inversion in absorptivities typical of metals compared to non-metals at very low wavelengths.

E. R. QUEER, State College, Pa.: I would like to direct your attention to the fact that colors may be very deceptive in the absorption characteristic. During the war we developed a camouflage type of paint that was very similar to this green. It was a dark green or tropical paint that had an absorbing coefficient of about 0.48. So the eye may be entirely misleading if you are trying to evaluate absorption characteristics. You must know the composition of the paint and in many instances you must have the measurements made to determine the coefficient.

C. S. LEOPOLD, Philadelphia: I wish to emphasize the importance of obtaining data for both the continuous spectrum and by wave lengths. In analyzing the path of radiant energy in an enclosure, it is necessary to have information that is related to wave length as each time radiation encounters a solid or gas, there is some absorption and this absorption is usually selective. An application of the grey body concept, based on coefficients obtained from the total spectrum, is frequently in error.

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* W. Sieber, *Zeitschrift für technische Physik*, 22, 1941, pp. 130-135.

J. D. Cannel, et al, showed that a white paint containing copper phosphate when exposed to the total spectrum of a filament lamp had a reflection factor of 30 to 35 percent, whereas the reflection factor of a white zinc oxide paint was 55 to 60 percent. I had occasion to confirm these values and, in addition, found that the reflection in the visible portion of the spectrum was reduced from approximately 83 to 75 percent. I believe that the analysis of light by wave lengths has engineering significance.

R. A. MILLER, Pittsburgh: I wondered about the authors' statement that Paint No. 7 was clearly white while Paints Nos. 5 and 6 were distinctly yellow in cast, noting that Paint No. 5 has only titanium oxide in the pigment which does not change in color with aging and remains permanently white whereas Paint No. 7 contains lithopone which is a lead compound material which does change in color and ages under the effect of sunlight. In Paint No. 6, titanium magnesium and zinc oxide are the principal components of the pigment but it again contains lithopone and again should be expected to change in color. If any of the three remained permanently white, I would expect this of the white enamel.

With reference to the table of effective coefficients, to amplify what Professor Queer said, the development of camouflage paint during the War was such that it was possible to produce colors that were much like the green of the objects you wished to camouflage. Actually they were ocularly camouflaged but not photographically. It was necessary to develop paints, the emissivity of which would be satisfactory as ocular greens and also satisfactory as photographic greens in order that the proper camouflage of those materials could be obtained. Consequently, the coefficient does not depend upon ocular appearance or upon the measurement of any one paint without positive knowledge of the components' values and the things that affect that absorption.

A. G. H. DIETZ, Cambridge, Mass.: The authors are to be congratulated for conducting this research on the effective solar absorption of colored paints. In the field of space heating by solar energy the absorptivities of various surfaces suitable for use in flat plate collectors or heat traps are of paramount importance. In such installations the maximum absorptivity is essential.

For solar heating installations in buildings the effect of time upon the collecting surface is important. These surfaces should undergo a minimum of change over a period of years. It would be valuable, therefore, to have information respecting the changes in exposure. The change noted in sample 6 after two months exposure is pertinent. Would the absorptivity increase over a much longer period? Would some of the surfaces which exhibit high absorptivities, such as sample 8, show a decrease over a period of years? Information of this kind is needed for the long-range prediction of the behavior of solar energy collectors.

G. V. PARMELEE, Cleveland: The authors are to be commended for the work which they have done on their own time in collecting data and preparing this paper. The data are of considerable value in estimating cooling loads.

In connection with the problem of reducing solar heat gain by shades, I have been very much interested in their work and in the values obtained. When the authors sent a copy of the paper to the Laboratory, I tried to estimate the effective absorption coefficient corresponding to assumed absolute absorption coefficients. The results of the calculations are given in Table A.

The true absorption coefficient, α, was used as the starting point. By solution of the heat balance equation the equilibrium temperature, t, was calculated for the various values of emissivity, e, of the sample surface seeing the sun and the convection coefficient, h_c, for upward heat loss. The air temperature was assumed to be 90 F and the incident solar radiation 200 Btu per (hr) (sq ft). Allowance was also made for low temperature radiation exchange between the sky and the sample. A value of 132 Btu per (hr) (sq ft) received by a horizontal surface from the sky was used in

TABLE A—CALCULATED EFFECTIVE SOLAR ABSORPTION COEFFICIENTS

ABSOLUTE SOLAR ABSORPTION COEFFICIENT α	BLACK HEAT EMISSIVITY e	CONVECTION COEFFICIENT h_c	EQUILIB. TEMP. DEG F t	EFFECTIVE SOLAR* ABSORPTION COEFFICIENT α_e
0.20	0.05	0.5	111.9	0.40
0.20	0.90		95.7	0.10
0.40	0.90		110.3	0.37
0.60	0.90		123.7	0.61
0.80	0.90		136.1	0.84
1.00	1.00		145.1	1.00
0.20	0.05	3.0	99.3	0.29
0.20	0.90		93.2	0.10
0.40	0.90		100.7	0.34
0.60	0.90		108.0	0.57
0.80	0.90		115.4	0.80
1.00	1.00		121.6	1.00

$$*\alpha_e = \frac{t - 90}{t_{\alpha = 1.0} - 90}$$

calculating this exchange. This is for 70 F dewpoint and is calculated from the equation due to Brunt†:

$$R_{ih} = R_{ib}\left(0.55 + 0.056\sqrt{M}\right)$$

where

R_{ih} = radiant energy received by a horizontal surface from the sky, Btu per (hour) (square foot).

R_{ib} = black body radiation at the temperature of the air, t, degrees Fahrenheit, at ground level.

$$= 0.173\ \frac{(t + 460)^4}{(100)}$$

M = vapor pressure, millibars, at ground level.

The under side of the sample was assumed to be a black body radiator. Convection downward was assumed to be zero.

The reason we are interested in the true coefficient is this: in analyzing the transmission of solar energy by a slatted blind, for example, a venetian blind, we find that the radiation hits the slat, bounces up and strikes the bottom of the slat above it and by successive reflections part of the radiant energy reaches the indoors. The important thing in this situation is the true or absolute coefficient, not the equilibrium temperature or effective coefficient.

It is interesting to note that for an absolute coefficient of 0.20 and an emissivity of 0.05, the effective coefficient is 0.40. This bears out the authors' statement that the true value for aluminum foil is considerably lower than the effective value which they have given. Note that if the black heat emissivity, e, were 0.90, the effective coefficient would become 0.10.

Note also that as the true coefficient increases, the effective coefficient becomes nearly equal to it. Much the same results are found when the convection conductance for the upper surface is increased to 3.0.

AUTHORS' CLOSURE: We are grateful for the comments on this paper, both oral and written.

Replying specifically to Mr. Stewart's discussion, this method of test would lend itself readily to all types of paints that could be applied to a thin sheet. The brick,

† Radiation in the Atmosphere, by D. Brunt (*Supplement to the Quarterly Journal of the Royal Meteorological Society*, Vol. 66, 1940).

marble, concrete and other common building materials would present more of a problem as they would have to be tested in comparatively thin sections in order to minimize the heat capacity effect, and variations in surface temperature due to differences in conductivity of the materials being tested. For an air to air temperature drop of 50 deg, the temperature drop through a 0.05 in. thickness of ordinary building brick would probably be less than 1 deg. This should allow sufficient precision in testing rigid materials. If these materials could not be ground or processed into thin sheets, we would probably have to rely on the coefficients already obtained by various observers at wavelengths of approximately 0.60 microns.

Mr. Bradley is quite right in his statement that at night a white roof will cool off more rapidly than an aluminum roof due to the high emissivity of the white roof.

Professor Queer has stated that the eye may be entirely misleading in trying to evaluate absorption coefficients. This is true, as has been observed by numerous investigators. The low absorption coefficient of 0.48 for the camouflage green paint is a striking example of this. It is unfortunate that we did not have any of this type of paint for test. Apparently the chemical composition of the paint is an important factor in determining its absorption coefficient. For instance, Coblentz obtained an absorption coefficient at 0.60 microns of 0.14 for White (ZnO_2) pigment and 0.26 for White (Y_2O_3) pigment. Mr. Leopold mentions the great differences in reflection factors of a white paint containing copper phosphate as against a white paint containing zinc oxide.

Mr. Miller evidently misunderstood our statement about the whiteness of the white paints. Sample 5 was supplied by the Research Laboratory and had a distinctly yellowish cast when it was received at the Institute, while sample 6 which was secured and applied to the aluminum foil by us, also had a distinctly yellowish cast when applied. These samples were not aged or exposed to sunlight before testing.

In regard to Professor Dietz's questions, we are of the opinion that the absorptivity of sample 6 would increase over a much longer period, and that the absorptivity of sample 8 would not decrease, but would probably increase over a period of time if the paint were not eroded away. A good grade of flat black paint should be excellent for solar energy collectors. However, if the paint is applied over a bright metal surface, for instance stainless steel, the paint would probably have to be renewed from time to time because, as the paint becomes thinner there will be a tendency for the absorption coefficient to approach somewhere between that of the paint and the stainless steel. Emissivity tests conducted some years ago with lampblack over polished silver indicated an emissivity coefficient of 0.70 for a lampblack thickness of 15 microns. When the thickness of the lampblack was increased to 72 microns, the emissivity increased to a value of 0.945.

We appreciate the interest of Mr. Parmelee and the time spent in making the calculations of effective coefficients for assumed absolute coefficients. These calculations confirm what we have noticed, namely that the true coefficient is probably much lower for aluminum foil than the effective coefficient, and the true and effective coefficients are very close to each other for true coefficients of 0.4 and higher.

We would like to mention, however, that it is impossible to calculate accurately the effective coefficient from an assumed absolute coefficient. For instance, if we assume the incident solar radiation at 240 Btu per (hr) (sq ft) instead of 200 Btu, and assume all the other variables the same as Mr. Parmelee has used in his calculations, the calculated effective coefficient for an assumed absolute coefficient of 0.40 is 0.39 instead of 0.37. For an incident solar radiation of 260 Btu (probably close to the actual value at the time of the tests), the calculated effective coefficient is the same as the assumed absolute coefficient or 0.40.

While Mr. Parmelee is interested in absolute coefficients in analyzing the transmission of solar energy by slatted blinds, the effective coefficients are probably close enough for all the materials tested, with the exception of the aluminum foil and

aluminum paint. The same is true for the designing engineer who is interested in the cooling or heating load.

In an attempt to get a correlation between the measured surface temperatures of the white and black painted samples, and calculated surface temperatures for thin sheets and insulated sheets exposed to solar radiation, heat transmission and surface temperatures were calculated for a thin sheet and for a 2-in. thick insulated sheet exposed to a solar radiation intensity of 200 Btu per (hr) (sq ft) with an air temperature of 90 F above the sheets and 80 F below the sheets, corresponding to a cooled tent and a cooled insulated slab having a *k* factor of 0.25. The results of the calculations are as follows:

Test Sample	a	ϵ	Upper Surface Temperature			Ratios	
			Test Sample	Thin Sheet	2 Inch Insulation	Test Sample / Thin Sheet	Test Sample / 2 Inch Insulation
White Enamel	0.42	0.9	104.9	104.5	113.5	1.004	0.924
Black Paint	0.95	0.95	139.6	133.1	150.2	1.049	0.929

The ratios of the temperatures of the test samples to that of thin sheets for absorption coefficients of 0.42 and 0.95 are within 4.5 percent of each other in spite of the fact that the air temperature on both sides of the test samples was approximately 74 F. In this case the film coefficients are the only resistances entering into the calculations, and there is some question as to the accuracy of Brunt's equation in calculating the reradiation from the sky. It is surprising to note that the ratios of the temperatures of the test samples to those of the insulated sheets are within 0.5 percent of each other. In most practical cases considerable insulation resistance would be present so that it seems reasonable to assume that heat transfer and surface temperatures could be calculated with a precision of one or two percent from the effective absorption coefficients, for most materials with the exception of polished metals and metallic paints.

Mr. Hooper intimates that the true solar absorption coefficient of aluminum foil is higher than that of white paint. This is questionable, also his assumption of 350 Btu per (hr) (sq ft) radiation intensity for Pennsylvania seems rather high and the equilibrium temperature of 100 F rather low.

It happens that tests were made on aluminum foil, white enamel and black paint within a comparatively short time of each other. From the constants of the radiometer and the known emissivity of the black painted surface exposed to the thermopile at all times it is possible to calculate the temperature of the thermopile and the three test samples. The calculated temperature of the thermopile was approximately 74 F for all three tests and the temperature of the black painted surface was 139.6, the white painted surface 104.9 and the aluminum foil 105.5 F. It is hardly possible that the absolute solar absorption coefficient of the aluminum foil could have been higher than that of the white paint; if it was, the surface temperature should have been much higher than the white paint, because it would have been absorbing more energy from the sun and owing to its very low *low temperature* emissivity it would necessarily radiate much less heat to the sky and both these factors would have a tendency to raise the surface temperature of the aluminum foil sample.

The selective radiation tests reported by W. Sieber in 1941 for aluminum and white fire clay at a wave length of 1 micron are contrary to the values obtained at the University of California in 1949 for aluminum foil and white enamel undercoater at the same wave length. The California tests gave an absorption coefficient of 0.07 for the aluminum foil and 0.49 for the white enamel undercoater, while the tests of Sieber show values of 0.22 for aluminum and 0.19 for white fireclay at temperatures of 5200 F absolute, as taken from the curve submitted by Mr. Hooper.

1401

No. 1401

A PROPOSED PSYCHROMETRIC CHART

By H. B. Nottage*, Cleveland, Ohio

This paper is the result of research carried on by The American Society of Heating and Ventilating Engineers at its Research Laboratory located at 7218 Euclid Ave., Cleveland 3, Ohio.

THE PSYCHROMETRIC chart is a fundamental tool of the air conditioning engineer. Acting upon the recommendation of the Guide Committee, the A.S.H.V.E. Committee on Research has sponsored a critical appraisal of psychrometric calculation principles and practices which has culminated in the preparation of a chart being presented to the Society in this paper. Technical guidance in this development has been given by the Subcommittee on Thermodynamics† of the Guide Committee.

In preparing the chart the subcommittee established the following requirements: (1) best available thermodynamic data should be used, (2) chart form and content to be consistent with indisputable thermodynamic principles, and (3) chart to be styled for convenient use.

Any psychrometric chart which is to be of real value must have distinctive features which aid in problem analysis. This is particularly true of the proposed chart, and guiding thermodynamic principles are emphasized in the discussion. Detailed consideration of all aspects of calculating techniques and the analysis of specialized types of problems however are beyond the scope of the present paper.

The Chart

Two versions of the chart have been suggested: (1) a large chart for precise calculations suitable for inclusion in The Guide, and (2) a smaller version about twice the size of Fig. 1, for more convenient use.

Also suggested are the use of more than one color and the addition of an auxiliary transparent protractor giving an alternative form of the ratio or slope scale.

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† T. F. Rockwell, *chairman;* H. D. Lifton, and H. B. Nottage.

Presented at the Semi-Annual Meeting of The American Society of Heating and Ventilating Engineers, Muskoka, Ontario, Canada, June 1950.

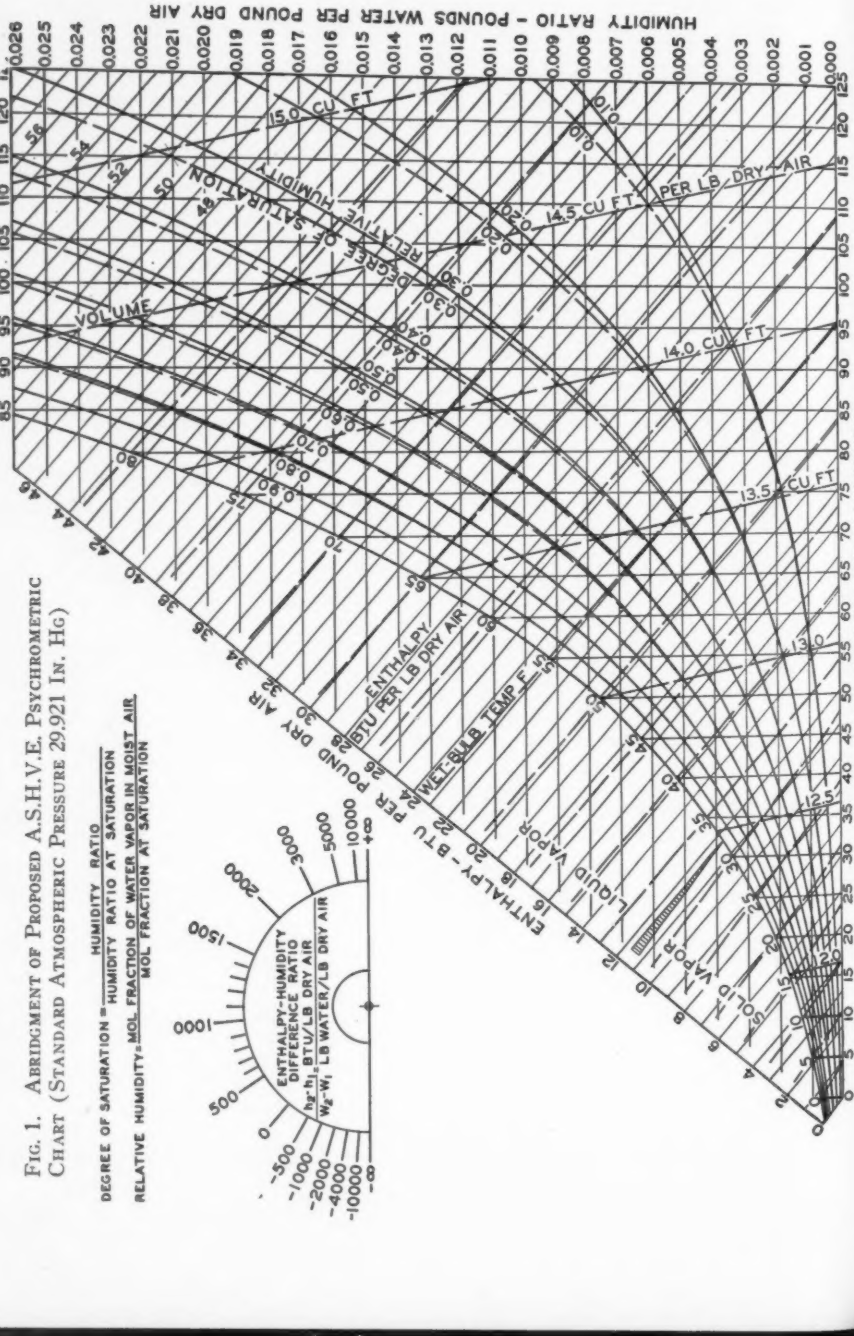


Fig. 1. Abridgment of Proposed A.S.H.V.E. Psychrometric Chart (Standard Atmospheric Pressure 29.921 In. Hg)

The chart is plotted on oblique coordinates of enthalpy and humidity ratio. This method of plotting was originated by Mollier[1, 2, 3] in 1923. Goodman[4] has followed this arrangement.

The chart is based upon the thermodynamic data of Goff and Gratch as given in THE GUIDE,[5] 1950; these data include effects of the interaction between air and water vapor. Quantities appearing on the chart are defined in THE GUIDE[5]

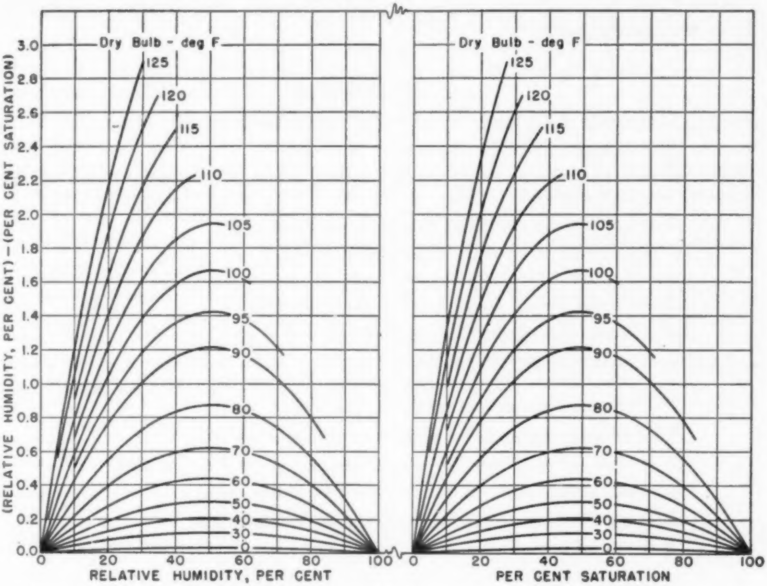


FIG. 2. COMPARISON OF RELATIVE HUMIDITY AND PERCENT SATURATION WITHIN THE RANGE OF THE CHART

except for the two defined explicitly on the chart itself, namely, relative humidity and degree of saturation.

Fig. 2 compares relative humidity and degree of saturation according to the identical relationship

$$R = \frac{s}{\left[1 - (1 - s)\, f_s \,\frac{P_s}{P}\right]} \quad \ldots\ldots\ldots\ldots \quad (1)$$

where

R = relative humidity, expressed as a decimal.
s = degree of saturation, expressed as a decimal.
P = observed (or barometric) pressure of the moist air.
P_s = saturation pressure of pure water at the prevailing temperature, expressed in the same units as P.
f_s = a dimensionless factor which may be regarded as accounting for influences arising when air and water are intermixed.

[1] Exponent numerals refer to References.

Example: Using the left diagram, at the intersection of 35 percent relative humidity and 90 F, the left hand scale indicates percent saturation as 1.1 *lower* than the relative humidity, or 33.9 percent. On the other hand, using the right diagram, at the intersection of 35 percent saturation and 95 F, the relative humidity is about 1.3 percent *higher* than the percent saturation, or 36.3 percent.

Magnitudes of f_s have been reported by Goff and Gratch[6] and by Goff[7]. Table 1 gives f_s for the conditions of the A.S.H.V.E. chart. The saturation pressure P_s is well established.[5, 7]

NOMENCLATURE

f_s = factor in Equation 1 and Table 1, dimensionless.

G = rate of flow of dry air, pounds per hour.

H = enthalpy of an *entire* non-flow system expressed as Btu per (pound of air in the system).

h = enthalpy of moist air, Btu per (pound dry air).

h_w = enthalpy of water, Btu per pound of water.
h_{wl} refers to liquid, h_{wv} to vapor, and h_{ws} to solid water, respectively.

L = rate of flow of liquid water, pounds per hour.

M_a = total mass of (dry) air within a non-flow system, pounds.

M_w = total mass of water within a non-flow system, pounds.
M_{wl} refers to liquid, M_{wv} to vapor, and M_{ws} to solid water, respectively.

P = observed pressure of moist air, psi or other units.

P_s = saturation pressure of pure water, psi or other units.

${}_1Q_2$ = heat added to non-flow system between states 1 and 2, Btu.

${}_1q_2$ = rate of heat addition to a flow system between states 1 and 2, Btu per hour.

R = relative humidity, defined on Fig. 1, dimensionless.

S = rate of flow of solid water, pounds per hour.

s = degree of saturation, defined on Fig. 1, dimensionless.

t_{wb} = thermodynamic wet bulb temperature, Fahrenheit degrees.

V = rate of flow of water vapor (pure, no air) pounds per hour.

w = humidity ratio of moist air, (pounds of vapor)/(pound of dry air).

TABLE 1—MAGNITUDES OF f_s FOR THE RANGE OF THE PROPOSED CHART
(Standard Barometric Pressure, 29.921 in. Hg)

TEMP F	f_s	TEMP F	f
0	1.0048	70	1.0045
10	1.0046	80	1.0047
20	1.0046	90	1.0048
30	1.0045	100	1.0050
40	1.0044	110	1.0053
50	1.0044	120	1.0055
60	1.0044	125	1.0057

Note: The original source[6] gives f_s to seven significant figures over the temperature range —208 F to +202 F and over the pressure range 20 to 35 in. Hg.

PRINCIPLES OF PSYCHROMETIC MASS AND ENERGY BALANCES

The foremost use of the psychrometric chart is in locating state points of moist air and in diagramming and calculating mass and energy balances for air conditioning equipment and systems. The principles of these balances are fundamental to load estimates, specifications, equipment selection, and performance analyses.

It is also possible to develop uses of the chart in treating point-to-point variations in the state of moist air within a process, as governed by the particular rates of heat and mass transfer existing between the overall limits of a change, but this latter topic is not considered in this paper.

Psychrometric mass and energy balances are restricted to include only the substances air and water (liquid, solid, or vapor), either as pure substances or as mixtures. The two most frequent kinds of balances are: (1) non-flow under constant pressure and (2) steady flow. Only these two cases will be treated in this paper. Furthermore, steady-flow systems treated here will be limited explicitly to those in which changes of kinetic and potential energies are negligible compared to enthalpy changes.

Within this limited category all balances may be most simply stated in terms of enthalpies, humidity ratios, heat and/or work quantities or rates, and mass flow rates or total masses. This observation is fundamental to the choice of enthalpy and humidity ratio as chart coordinates. Non-flow and steady-flow balances are summarized in following paragraphs.

1. *Non-flow balances at constant pressure:*

(a) Mass balance

$$M_w + M_a = C \quad \text{(2)}$$

$$\frac{M_w}{M_a} = \frac{C}{M_a} - 1 \quad \text{(2a)}$$

where

M_w = mass of water (liquid, solid, vapor) in the system, pounds.
M_a = mass of air (dry) in the system, pounds.
C = a constant.

If only the gaseous phase is involved, Equation 2a specifies a constant humidity ratio.

(b) Energy balance ($M_a \neq 0$).

$$\frac{{}_1Q_2}{M_a} = H_2 - H_1 \quad \text{(3)}$$

where

${}_1Q_2$ = heat added to the system between states 1 and 2, Btu.
H = enthalpy of the *entire* system expressed as Btu per (pound of air in the system).

In the majority of air conditioning problems it is satisfactory to neglect the solubility of air in liquid and solid water. The enthalpy then is expressed as Btu per pound of air in the gaseous phase, or, more conventionally, as Btu per pound of dry air. Also, with liquid and/or solid water present, a further convenience is to resolve the enthalpy of the entire system into constituent parts according to phase, *i.e.*,

$$H_2 - H_1 = \left(h + \frac{M_{wl}}{M_a} h_{wl} + \frac{M_{ws}}{M_a} h_{ws}\right)_2 - \left(h + \frac{M_{wl}}{M_a} h_{wl} + \frac{M_{ws}}{M_a} h_{ws}\right)_1 \quad . \quad (4)$$

where

h = enthalpy of ***moist air***, Btu per pound of dry air.
h_{wl} = enthalpy of liquid water, Btu per pound.
h_{ws} = enthalpy of solid water, Btu per pound.
M_{wl} = mass of liquid water, pounds.
M_{ws} = mass of solid water, pounds.

With air solubility neglected, h_{wl} and h_{ws} are taken for the pure substance.
If only the gaseous phase is involved, Equations 2 and 3 yield an enthalpy change at constant humidity ratio.

2. *Steady-flow balances:* Two possibilities in general may be noted: (1) total rate of mass inflow for all streams entering a system equal to total rate of mass outflow for all streams leaving, and (2) total mass inflow rate not equal to total mass outflow rate because of some substance being deposited or picked up within a system which may not be directly discernible by an observer outside of the system. The deposition or sublimation of ice on the surface of an enclosed coil is an example of the second possibility. Only the first case however will be given detailed discussion here.
The solubility of air in liquid or solid water is to be neglected for present purposes.

(a) Mass balance.

$$\sum_{\text{inflow}}[G\ (1 + w) + L + V + S] = \sum_{\text{outflow}}[G\ (1 + w) + L + V + S] \quad . \ . \ . \quad (5)$$

where

G = flow rate of ***dry air***, pounds per hour.
w = humidity ratio, (pounds water vapor) per (pound dry air).
L = flow rate of liquid water, pounds per hour.
V = flow rate of water vapor (pure, no air in stream), pounds per hour.
S = flow rate of solid water, pounds per hour.
$\sum$ = denotes summation over all inflow or outflow streams.

(b) Energy balance.

$$\sum_{\text{inflow}}(Gh + Lh_{wl} + Vh_{wv} + Sh_{ws}) + q_i + W_i =$$

$$\sum_{\text{outflow}}(Gh + Lh_{wl} + Vh_{wv} + Sh_{ws}) \quad . \ . \ . \ . \ . \ . \ . \ . \ . \ . \quad (6)$$

where

h_{wv} = enthalpy of water vapor (pure, no air in stream), Btu per pound.
q_i = rate of heat inflow, Btu per hour.
W_i = rate of work input, Btu per hour.

While it is true that the relationships given introduce no properties beyond enthalpy and humidity ratio, practical utility of the chart as a universal tool calls for the inclusion of dry bulb and wet bulb temperatures, volumes and indices

of the condition of the air in relation to saturation. These properties all are needed in locating and describing states on the chart.

ELEMENTS OF PSYCHROMETRIC MASS AND ENERGY BALANCE CALCULATIONS

Mass and energy balances deal only with net changes between definite states; the detailed history of a change is not involved. A chart used to facilitate such

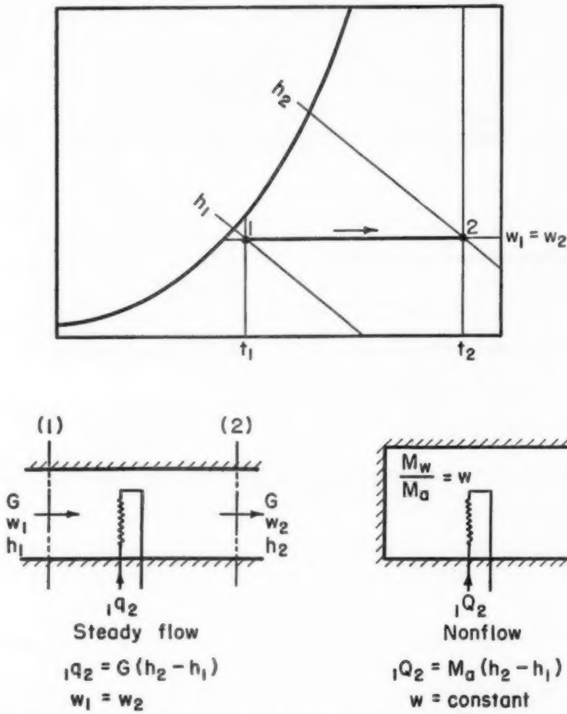
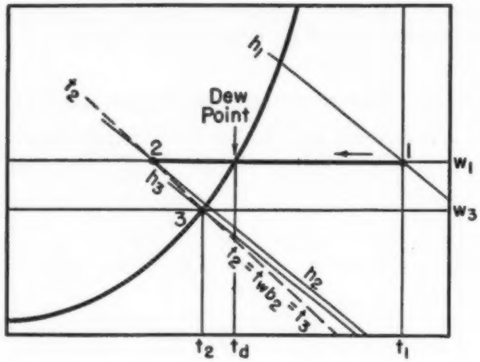
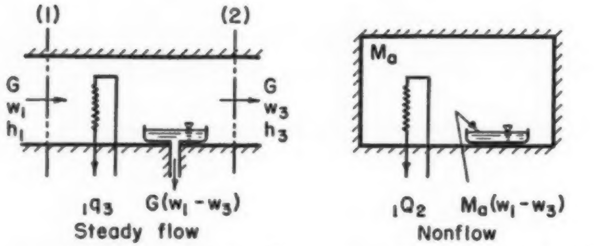


FIG. 3. HEATING OF MOIST AIR AT CONSTANT PRESSURE

calculations must primarily aid in clearly establishing states. Lines drawn on the chart to connect different states need have no other significance than being loci lines, that is, lines which contain the two terminal points of a change according to particular overall conditions imposed. Loci lines are commonly called *condition lines* for the processes concerned. On the proposed chart a condition line is characterized by its slope, $(h_2-h_1)/(w_2-w_1)$.

Use of the chart of Fig. 1 is limited to standard barometric pressure. Steady-flow changes commonly involve pressure drops with the flow, but so long as





$${}_1q_3 = G\left[h_1 - h_3 - (w_1 - w_3)h_{w_3}\right] \qquad {}_1Q_2 = M_a\left[h_1 - h_3 - (w_1 - w_3)h_{w_3}\right]$$

$${}_1q_2 = G(h_1 - h_2) = {}_1q_3 \qquad = M_a(h_1 - h_2)$$

FIG. 4. COOLING OF MOIST AIR AT CONSTANT PRESSURE WITH CONDENSATION OF LIQUID WATER

these pressure drops remain a small fraction of the barometric pressure no appreciable errors need arise from using a constant-pressure chart.

Simple examples are given in the following paragraphs 1 to 7 to illustrate the elements of psychrometric calculations in terms of enthalpies, humidity ratios, and mass flow rates or total masses. In all instances the solubility of air in liquid or solid water is neglected.

1. Fig. 3, heating of moist air at constant pressure.
2. Fig. 4, cooling of moist air at constant pressure with condensation of liquid water.

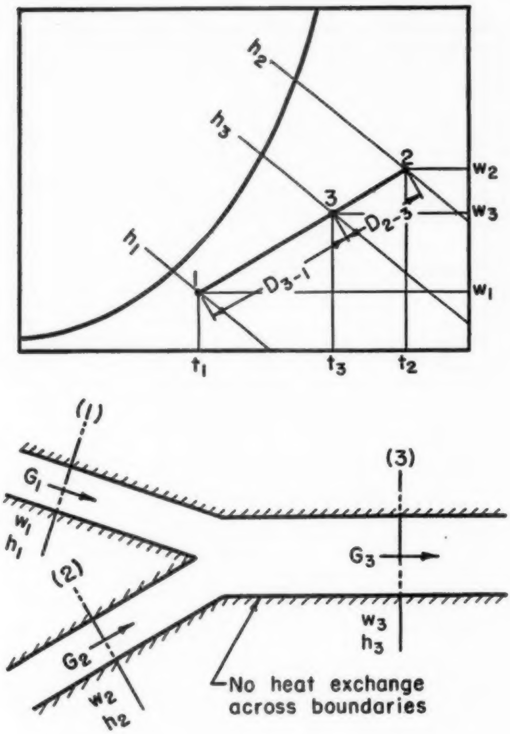
By definition the dew point corresponding to any state such as 1 is t_d as shown.

The enthalpy h_{w3} is that of liquid water at temperature $t_3 = t_2$ and under barometric pressure.

This example illustrates one use of the chart region to the left of the saturation line. It is important to recognize that $G\ (h_1 - h_3) \neq {}_1q_3$.

3. Fig. 5, adiabatic mixing of two steady-flow streams of moist air at constant pressure.

The distances $D_{3\text{-}1}$ and $D_{2\text{-}3}$ are scaled linearly from the chart.



$$G_1 + G_2 = G_3 \qquad G_1 w_1 + G_2 w_2 = G_3 w_3$$

$$G_1 h_1 + G_2 h_2 = G_3 h_3$$

$$\frac{G_1}{G_2} = \frac{h_2 - h_3}{h_3 - h_1} = \frac{w_2 - w_3}{w_3 - w_1} = \frac{D_{2-3}}{D_{3-1}}$$

FIG. 5. ADIABATIC MIXING OF TWO STEADY-FLOW STREAMS OF MOIST AIR AT CONSTANT PRESSURE

A convenient form in which to remember the mixing rule is $G_1D_{3-1} = G_2D_{2-3}$.

4. Fig. 6, addition of moisture to an adiabatic system in steady flow at constant pressure.

The moisture may be added at any state (liquid, solid, vapor) but all of the moisture must be in the vapor state at section 2 for the system visualized. For one example, a proper choice of h_w will yield a substantially isothermal change of humidity ratio.

5. General adiabatic saturation at constant pressure.

Adiabatic saturation denotes any process wherein the state of moist air is changed from some initial condition to the saturation curve in an adiabatic manner. The

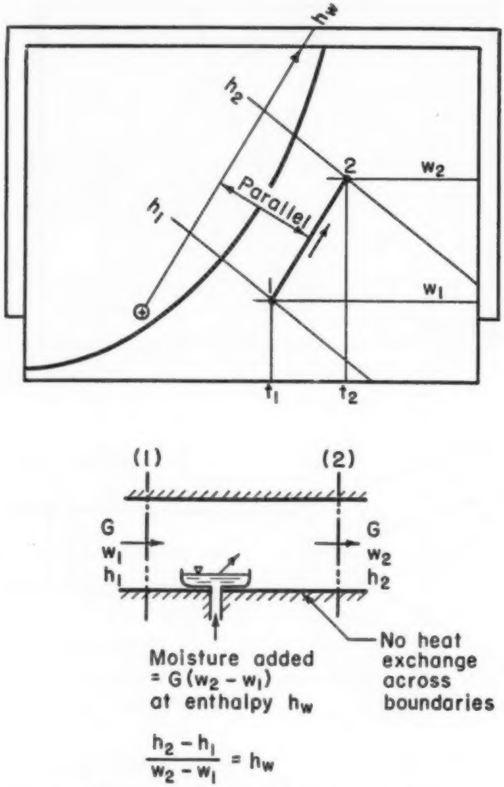


FIG. 6. ADDITION OF MOISTURE TO AN ADIABATIC SYSTEM IN STEADY FLOW AT CONSTANT PRESSURE

moisture addition dealt with in Fig. 6 may become adiabatic saturation within this definition if the line 1-2 intersects the saturation curve.

6. Fig. 7, adiabatic saturation or adiabatic humidification at constant pressure with recirculated spray water and steady flow. In a saturation process the spray water ultimately will come to the same temperature as the saturated leaving air; this temperature is, by definition, the thermodynamic wet bulb temperature, t_{wb}. The adiabatic *saturation* process has its final state on the saturation curve at the wet bulb temperature of the entering air, while an adiabatic *humidification* process is one in which $t_{wb2} = t_{wb1}$ but for which the final state does not reach full saturation.

7. Fig. 8, exchange of heat and moisture between air and water streams in steady flow at constant pressure. Given state 1, the ratio $(h_2 - h_1)/(w_2 - w_1)$ establishes the slope of the *condition line* which must contain state 2. Setting $L_2 = O$ yields the case of heat and moisture addition without through flow of water.

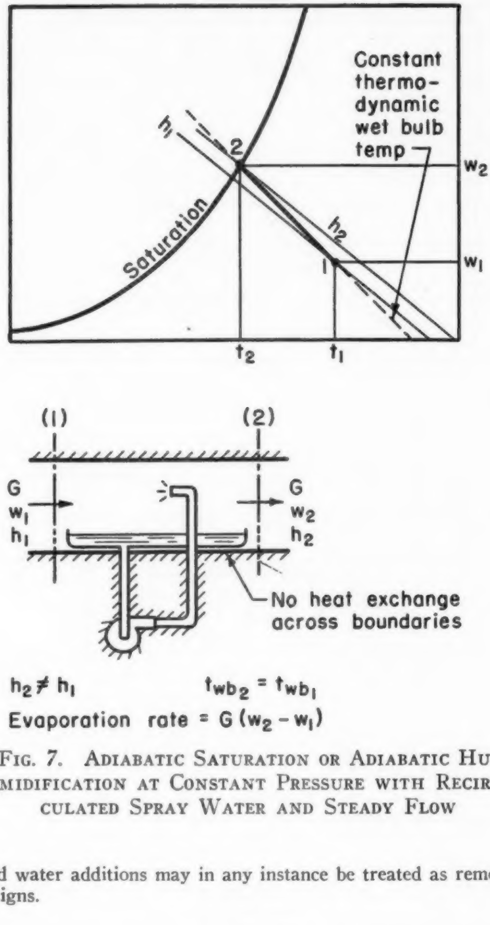
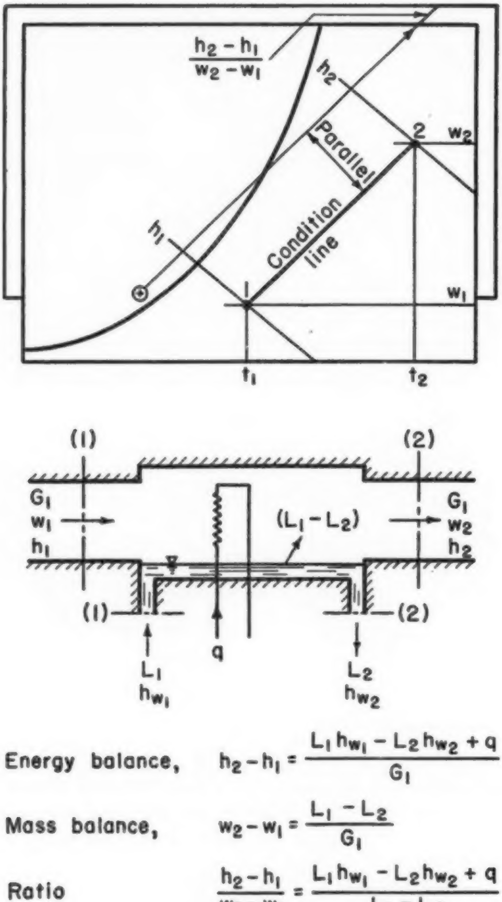


FIG. 7. ADIABATIC SATURATION OR ADIABATIC HUMIDIFICATION AT CONSTANT PRESSURE WITH RECIRCULATED SPRAY WATER AND STEADY FLOW

The heat and water additions may in any instance be treated as removals by changing algebraic signs.

DISCUSSION

In becoming familiar with energy-balance calculations the following sequence of steps in problem solving is suggested: (1) a schematic diagram of the system concerned, noting known and unknown quantities, (2) a sketch of the psychrometric chart upon which a skeleton layout and solution of the problem may be developed, (3) the writing of mass and energy balances, and (4) expression of condition-line slopes in terms of known or determinable quantities. Balances may be written for the system as a whole or for any part which may be schematically isolated, *e.g.,* a coil, a mixing section, a washer, a fan, a room, a dehumidifier, or a combination of appropriate elements.



Energy balance, $$h_2 - h_1 = \frac{L_1 h_{w_1} - L_2 h_{w_2} + q}{G_1}$$

Mass balance, $$w_2 - w_1 = \frac{L_1 - L_2}{G_1}$$

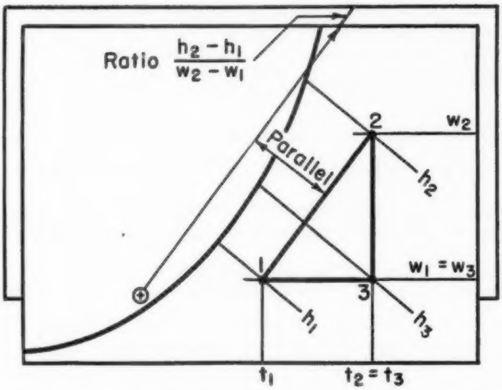
Ratio $$\frac{h_2 - h_1}{w_2 - w_1} = \frac{L_1 h_{w_1} - L_2 h_{w_2} + q}{L_1 - L_2}$$

FIG. 8. EXCHANGE OF HEAT AND MOISTURE BETWEEN AIR AND WATER STREAMS IN STEADY FLOW AT CONSTANT PRESSURE

Psychrometric literature often refers to the "sensible" and "latent" components[8] of a moist-air enthalpy difference. Since the enthalpy difference between any two states is independent of the actual process path a difference such as $(h_2 - h_1)$ in Fig. 9 may be expressed as $(h_2 - h_3) + (h_3 - h_1)$, wherein point 3 is located by the constant humidity ratio line through 1 and the constant dry bulb line through 2. The quantity $(h_2 - h_3)$ may be called a "latent" difference and $(h_3 - h_1)$ a "sensible" difference if desired. If a triangle such as 1-2-3 main-

tained the same dimensions in all regions of the chart for a fixed length and slope of the side 1-2, then the sides 1-3 and 2-3 could serve to establish the remaining side in any location. This condition however is not true on the proposed chart, a triangle such as 1-2-3 is not in general a right triangle and hence slopes of lines such as 1-2 are expressed directly as $(h_2-h_1)/(w_2-w_1)$. An adjustment of practical usage is desirable to characterize a condition line in terms of the "air enthalpy load," (h_2-h_1), and the "moisture load," (w_2-w_1).

The use of enthalpy and humidity ratio as coordinates gives a chart which can readily be extended to other temperature ranges and to other barometric



Note: Angle 1-3-2 is exactly 90 degrees only for the 70 F dry bulb temperature line for the chart of Fig. 1.

FIG. 9. ISOTHERMAL AND CONSTANT-HUMIDITY-RATIO COMPONENTS OF AN ENTHALPY CHANGE

pressures. The mass and energy balance calculations as formulated algebraically are not restricted as to temperature and pressure.

CONCLUSIONS

1. The proposed chart, plotted on coordinates of enthalpy and humidity ratio and reproduced in two colors as recommended, offers thermodynamic consistency and convenient styling with no radical departure from other currently-used charts plotted on coordinates of humidity ratio and dry bulb temperature.

2. Mass and energy balance calculations employing enthalpy and humidity ratio are in accord with indisputable thermodynamic principles and are best organized with reference to a chart employing these properties as coordinates.

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APPENDIX

DEGREE OF SATURATION AND RELATIVE HUMIDITY

The question of whether the proposed chart should contain percent saturation, relative humidity, or both, has been the most controversial single item considered by

CONSIDERATION	RELATIVE HUMIDITY	DEGREE OF SATURATION
1. Definition	Mol fraction ratio is definite and applicable to real gas mixtures. Past definitions introducing perfect gas hypotheses have been inexact	Humidity-ratio ratio is definite and applicable to real gas mixtures. Can be applied to perfect gas idealization
2. Units	Molal units not generally employed in air conditioning	Pound units are familiar
3. Past Usage in Reporting Data	Large body of information expressed in relative humidity terms	Limited usage
4. Comparative Magnitude	Both quantities same order of magnitude in range of Chart; see Fig. 2	
5. Aid in Tabular Calculations	Impractical; requires conversion to pound units for use of tables in pound units. Would find direct use, linear interpolation in chart range, if tables were in molal units	Direct use, linear interpolation in chart range, for tables in pound units. Impractical if tables were in molal units
6. Use in Mass Transfer Rate	Usable with advantage that molal units are more fundamental for diffusion calculations and most directly applicable to thermodynamic equilibrium relationships	Usable for most rate calculations in range of chart. Convenient for introduction of Lewis relationship
7. Application to Perfect Gas Hypothesis and Dalton's Rule	Simplest in definition of *partial pressure.* As an idealization f_s in Equation A-1 may be taken as unity	Applicable through what amounts to a conversion to molal units

the Subcommittee. Presentation of both quantities on Fig. 1 of the paper is a compromise. The preferred styling would be to have only the most convenient and useful of the two quantities shown on the chart and then employ the curves of Fig. 2 as an interrelationship.

The explicit statements on Fig. 1 define the two quantities under consideration. The definition of relative humidity has been chosen to avoid any sort of ambiguity arising through a Dalton's rule idealization. It is still possible to *define a partial pressure* as the product of mol fraction and total pressure; proceeding therefrom, the relative humidity definition may alternatively be written

$$R = \frac{x_w P}{x_{ws} P} = \frac{x_w P}{f_s P_s} \quad \ldots\ldots\ldots\ldots\ldots \quad \text{(A-1)}$$

where

x_w = mol fraction of water vapor in moist air at the prevailing pressure and temperature, dimensionless.

x_{ws} = x_w at saturation for the same pressure and temperature, dimensionless.

P, P_s, f_s are defined in paper following Equation 1.

Judgment of convenience and utility between percent saturation and relative humidity requires thorough consideration. The accompanying tabulation summarizes many of the factors.

SUMMARY

1. The weight of past usage, even though with inexact definition, favors relative humidity. This may be modified however by item 4 in the outline which would permit interchangeability for many practical cases.

2. So long as the air conditioning engineer deals in pound units, percent saturation has a strong advantage for calculation use.

3. If the air conditioning engineer were to convert to molal units, then the case for percent saturation would essentially collapse.

4. Neither quantity as defined has physical meaning beyond the range of states wherein the liquid and vapor phases may coexist in neutral equilibrium. If only the vapor phase is possible, the preference for molal or pound units, or a fractional or ratio composition would decide the issue and the recommended terminology would be either: (1) mol fraction, (mols) per (mol of mixture), (2) mol ratio, (mols) per (mol of other component), (3) mass fraction, (lb) per (lb mixture), or (4) mixing ratio, (lb) per (lb of other component).

DISCUSSION

C. M. ASHLEY, Syracuse, N. Y. (WRITTEN): The proposed psychrometric chart presented by the author represents an important advance in understandability and usability over the present A.S.H.V.E. psychrometric chart. Whether this chart will gain general acceptance in competition with existing charts having less confusion of nearly parallel lines remains for future determination. I would question whether the chart still has attained the proper degree of usability but perhaps judgment should be reserved on this until it is possible to work with the chart in a larger size and in two colors.

It is of interest to note that the enthalpy-humidity difference ratio $h_2 - h_1/w_2 - w_1$

is directly related to sensible heat factor as used in my paper* in a relatively simple manner by the equation:

$$\text{Sensible heat factor} = 1 - \frac{1076}{\frac{h_1}{w_1} - \frac{h_2}{w_2} - 20}$$

In connection with the author's discussion of *sensible* and *latent* heat components whose sum makes up the *total* heat, it is of some significance to note that the total heat is not $h_1 - h_2$ but rather it is $h_1 - h_2 + (L_1 h_{w_1} - L_2 h_{w_2})$. As has been shown, this value is better expressed in terms of the sum of the sensible and latent components as related to temperature and humidity ratio difference respectively than in terms of $h_1 - h_2$ alone.

This chart has the advantage that a mixture condition lies on the straight line joining the two initial state points, whereas on the humidity ratio-dry bulb temperature chart it does not do so precisely. However, the deviation from a straight line on the other type of chart is extremely slight and is almost never of practical consequence.

Of particular interest is the comparison of percent saturation and relative humidity. I would like to make a strong plea for the use of relative humidity rather than percent saturation for the following reasons:

1. Relative humidity expresses the physical relationship of vapor pressure at the state point to that at saturation virtually independent of barometric pressure.

2. Relative humidity can be used above the boiling temperature of water, whereas percent saturation cannot. Also, it does have a definite physical meaning beyond this range since it is perfectly possible to describe the pressure at saturation in this region in terms of the vapor pressure of water. It should be noted that these deficiencies of percent saturation would be particularly embarrassing in drawing psychrometric charts at higher temperatures and for other barometric pressures.

3. Relative humidity serves as a good index of the moisture content of hygroscopic materials and, unlike the relationship to percent saturation, the relative humidity-moisture content relationship is virtually unaffected by barometric pressure and can be used in the range above the boiling point of water.

4. Relative humidity has a broad acceptance and many properties are defined in terms of it.

5. Percent saturation can always be obtained by a simple computation directly from the chart, whereas relative humidity cannot. I would not for a minute argue against the utility of percent saturation in the solution of certain types of problems, but believe that it can be adequately used without having to express its values specifically on the psychrometric chart.

R. D. WOOD, Syracuse, N. Y. (WRITTEN): I would like to offer two brief comments:

1. It is believed that the final chart would have much greater acceptance by air conditioning engineers if relative humidity, rather than percent saturation, is used.

2. From the standpoint of legibility, any chart showing both enthalpy and wet bulb lines should not only be in two colors, but should be larger than 11 × 17 in. In other words, it should be more than twice the size of the chart shown in Mr. Nottage's paper.

Although the proposed chart is an improvement over the one now used in THE GUIDE, it is believed that there should be no compromise with legibility or readability if the final chart is to gain wide acceptance.

DAVID LIFTON, State College, Pa. (WRITTEN): Mr. Nottage has done an excellent job of presenting the case for the proposed psychrometric chart. The Working Subcommittee, International Joint Committee on Psychrometric Data is in complete

* Psychrometric Factors in the Air Conditioning Estimate, by C. M. Ashley (A.S.H.V.E. TRANSACTIONS, Vol. 55, 1949, p. 91).

agreement as to the desirability of a new chart, so perhaps additional emphasis on some of Mr. Nottage's statements will be in order.

In a question such as this, personal experience with one type of chart will undoubtedly influence one's opinion concerning another type. Actually, the proposed chart does not depart radically from other commonly used charts, so that methods of using it would not be much different from current methods with other charts. It does, however, have certain advantages, the chief ones being:

1. The proposed chart is based on the best available data on the thermodynamic properties of air-water vapor mixtures instead of properties obtained from the perfect gas relationships or modifications of these relationships.
2. True values of enthalpy for any state on the proposed chart may be read directly instead of reading enthalpy at saturation, total heat or some other thermodynamically inconsistent property. Unfortunately, lines of constant enthalpy very closely follow lines of thermodynamic wet bulb temperature and this could lead to confusion if the chart were printed in one color, so it has been proposed to publish the chart in two colors. The problems involved in reproducing such a chart are being investigated.

The proposed chart is the first one using accurate data and incorporating lines of constant enthalpy in a simple, readily usable form. As such, it represents a significant advance.

T. E. GRAVENSTRETER*, Cleveland (WRITTEN): The author and the Society are to be congratulated for undertaking and bringing to a very practical conclusion a basic usable chart for the industry.

I have always felt that any information of value in the field should have for its source a non-commercial institution. Too much data in air conditioning has been compiled by firms which manufacture equipment. In this chart we have an original basic tool for the air conditioning engineer, and yet there are no popular trade names appearing on it. This work should do much to elevate the Society in its field of heating, ventilating and air conditioning.

S. H. DOWNS, Kalamazoo: I would like to ask Mr. Nottage a question. In Mr. Ashley's comments he referred to varying barometric pressures but I do not think he brought out strongly enough that we have to look into that subject. Perhaps Mr. Nottage already has a plan to give us a way to figure these values for other pressures than standard barometric pressures because I know of practically no installations that go in at standard barometric pressures. You either have an elevation to consider or some other conditions. Also, what are the plans to cover the range of dry bulbs higher than 125 F shown on the chart.

LESTER T. AVERY, Cleveland: It might be of interest to know that the preparation of this new Psychrometric Chart was discussed two years ago and authorized a year ago. The charts that we are using have trade names on them and we did not feel that we wanted to continue always depending on commercial psychrometric charts. Another point was that the information on the existing charts was not considered satisfactory. So we authorized the Laboratory to undertake this investigation with Mr. Nottage, and this is the result of it. Those of us who use charts will like it when we become familiar with it. I would like to congratulate Mr. Nottage on it.

C. S. LEOPOLD, Philadelphia: Mr. Nottage attended the last meeting of the International Joint Committee on Psychrometric Data, of which Dr. Wood is the Chairman and which I attended as the *ASRE* representative. The concepts on which the chart is based and the method of presenting the data were favorably commented on by the entire committee, including Dr. Goff.

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* Development Engineer, Bryant Heater Division.

RAYMOND MANCHA, Pittsburgh: It seems to me that anyone who is going to apply himself sufficiently to be able to intelligently use any chart in any of the existing forms, might just as well master the right chart without any more mental effort. In other words, I think there are many people who have been using charts who do not really know what enthalpy is. But they should learn it and it is not much harder to learn it the right way than a compromise way. I think among practicing engineers there are many who are not too familiar with existing charts.

W. R. HEATH, Buffalo: I would like to compliment Mr. Nottage on doing a very excellent job that certainly needed to be done. By all means we should have a standard chart sponsored by the A.S.H.V.E. and we should not have trade names on it. I also believe it should be complete and highly accurate.

I would like to put in a plea, however, for the working engineer who does 90 percent of the work with the psychrometric chart and does not need a high degree of accuracy. He does need something easy to use.

I would suggest, if it is still the intention to publish this in a pad form, that the chart include either enthalpy or total heat lines but not both, merely as a means of simplification, and that it be streamlined just as much as possible, even to the extent of sacrificing some accuracy. My thought is that the scientist or the research man requiring a high degree of accuracy can go to this large multi-colored chart, but that the working engineer has a simplified pad in front of him that he can use with reasonable accuracy, can tear a sheet off and file it with the estimate if he wishes.

Mr. Nottage mentioned *problem-solving technique,* and I would think that by all means a little presentation of that would be very valuable to all of us before we could make an intelligent decision as to whether we like the chart or not.

JOHN EVERETTS, JR., Philadelphia: There is no question in my mind but what the Society needs a new chart. However, if we are going to have a new chart and one that is distinctive, I would like to see some things added. I am quite sure that they can be added without involving confusion.

More and more we are getting into problems which involve variations in altitude. I notice that the chart that Mr. Nottage has worked up is based primarily on sea-level conditions. The only chart I know of that does take into account variations in altitude is one put out by the U. S. Bureau of Mines in which you can obtain any psychrometric conditions from 8000 ft below sea level to an altitude of 8000 ft above sea level. I know we have had problems in which we are interested in altitudes up in the neighborhood of 7300 ft and have had to redesign and develop a psychrometric chart for that purpose. I would like to ask Mr. Nottage if he could or would contemplate including altitudes other than sea level?

R. D. MADISON, Buffalo: I wish to compliment Mr. Nottage upon the development of this chart and I think the proof of its usefulness will be shown over a period of time when we can use the chart and find out what it actually will do for us as an engineering tool. I have had occasion to try to explain some of the old charts to various people and find it somewhat difficult and I know it is going to be quite difficult to explain this chart to a large field of engineers.

I saw a reference in the text to dew point and I think some reference should be made to dew point on the chart because it is one of the things we must use. I would like to put in a plea for a simpler chart for the practicing engineer—whether it is enthalpy or total heat, so that he can master one or the other and have reference to the complete chart which will probably appear in THE GUIDE.

CYRIL TASKER, Cleveland: President Avery has told you that the paper presented by Mr. Nottage is the culmination of about two years of discussion and study. Most of the real work on the problem was concentrated in the final three or four months,

and the many discussions held during the previous 18 months made the work of the last three or four months possible.

At the Laboratory we have operated on this project under the Guide Committee who requested the Committee on Research to arrange that the Laboratory staff could undertake this work. In presenting the chart to the Society at this time, we are therefore carrying out an assignment of one of the major Society committees whose job it is to present data in such a form that they can be used by the practicing engineer. The Laboratory has presented this chart, sponsored by a subcommittee of the Guide Committee. I would like to compliment Mr. Nottage on the excellent way in which he has carried out this assignment and on the way in which he has presented the material to you.

AUTHOR'S CLOSURE: The helpful comments received are greatly appreciated. If the proposed chart is adopted by the A.S.H.V.E., its purpose will be to best serve the membership and the profession. In this sense the chart will belong to the users, and its success will rest with them.

Use to maximum advantage calls for a whole-hearted adoption of enthalpy and humidity ratio in practical calculations for energy and mass balances. This may require changes in some contemporary practices, and these changes would require careful consideration.

An important hope is that the chart and the thermodynamic considerations which it embodies will be given immediate careful attention by the teaching profession. As remarked by Mr. Mancha, it is easy to begin by learning the fundamentals thoroughly. Skillful approximations, important in the practical art, then may be introduced on a clear and simple basis.

Fig. 1 of the paper is quite inadequate to portray the chart in the form visualized for convenient use. Advice from persons skilled in high-precision lithography has been that the combined use of color, very careful distinctions in line weight, and dashed *vs.* solid lines will produce a pleasing and easily-read chart.

Matters brought out by the discussers are conveniently grouped by topic for reply.

1. Other Pressures and Temperatures: The broad project initially contemplated recognized the desirability of covering an extended range of temperatures and pressures. The present chart is merely the beginning. Additional steps depend upon relative need and authorization. Coordinates of enthalpy and humidity ratio are well adaptable to other ranges.

2. Calculating Techniques: A comprehensively detailed study of calculating techniques, time-saving procedures, accuracy, and systematic convenience is lacking as a basis for comparing charts and computing methods. This was part of the initial broad considerations, but circumstances have caused postponement. The examples in the paper are highly skeletonized but nevertheless exemplify the general approach to practical problems.

3. Enthalpy and the Sigma Function: The original diagram of Mollier did not contain lines of thermodynamic wet bulb temperature. Instead, a procedure was given for drawing these lines, as needed, from the ratio or slope scale.

The strong traditional position of wet bulb temperature in American literature and practice necessitated inclusion on the proposed chart. The wet bulb lines also establish magnitude of the sigma function (sometimes called total heat). A frequent calculating procedure has been to replace changes in enthalpy, the fundamentally correct quantity, by changes in the sigma function. Adoption of enthalpy as a chart coordinate is intended to favor its direct use, for this makes it easy to read enthalpy directly at any point.

Possible confusion between the wet bulb and enthalpy lines is intended to be avoided by careful use of color, line weight, and dashed *vs.* solid lines. The line density could be reduced by drawing the wet bulb lines at, say, intervals of 2 degrees Fahrenheit.

4. Relative Humidity and Percent Saturation: Relative humidity, by long tradition, has a firm position in literature and practice, albeit frequently with inadequate definition. The question of establishing the best definition is under consideration by the International Joint Committee on Psychrometric Data, and it is hoped that their

decision will be generally adopted. It is expected that this definition will include the region above the boiling point of water.

Percent saturation is more convenient than relative humidity when using tabular thermodynamic quantities based upon the pound as the unit of mass. This is the major reason for its inclusion on the chart, although any small-size chart which might be printed probably would be little used with tabular calculations and hence percent saturation might be omitted in that instance. Availability of Fig. 2 aids interconversion and also would favor reducing the line density on a small-size chart by omitting percent saturation.

Matters concerning the possible printing and distribution of the proposed chart are without the province of the A.S.H.V.E. Research Laboratory. Thus, the suggestion represented in preliminary form by Fig. 1 is subject to alteration.

1402

No. 1402

A SIMPLE HEATED-THERMOCOUPLE ANEMOMETER

By H. B. Nottage*, Cleveland, Ohio

This paper is the result of research carried on by The American Society of Heating and Ventilating Engineers at its Research Laboratory located at 7218 Euclid Ave., Cleveland 3, Ohio.

A SIMPLE yet reliable instrument for measuring low air velocity is essential in research on air distribution in rooms. The heated-thermocouple anemometer is one type of remote-reading instrument which has shown promise for obtaining time-mean velocity data. Preliminary experience is reported here for units which have been built and calibrated at the A.S.H.V.E. Research Laboratory for the velocity range 30 to 1500 fpm. These instruments are currently being used in the long-range program[1, 2] on room air distribution under guidance of the Technical Advisory Committee on Air Distribution†. Inherent flexibility of design and application of the anemometers, together with inexpensive construction and adaptability to standard indicating or recording devices, are factors which favor diverse practical use of such instruments.

Heated-Thermocouple Principles and Theory

The temperature difference between an electrically heated thermo-junction and a differentially-connected unheated thermo-junction similarly placed in a fluid stream can be related to the velocity by calibration[3, 4]. Some commercially available anemometers employ this principle. One non-directional variation developed initially for cold-storage research employs the transient cooling of a small heated sphere[5].

The anemometer described in this paper is shown diagrammatically in Fig. 1. The small streamlined tip contains an electrical heating element wound about a thermocouple junction. The unheated junction is exposed directly to the air

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[1] Exponent numerals refer to References.

† Personnel: Ernest Szekely, *chairman;* N. E. Berry, H. F. Brinen, R. M. Conner, S. H. Downs, Linn Helander, F. B. Holgate, F. J. Kurth, J. N. Livermore, R. D. Madison, G. E. McElroy, L. G. Miller, D. W. Nelson, W. A. Pownall, G. B. Priester, C. H. Randolph, T. H. Troller, G. L. Tuve.

Presented at the Semi-Annual Meeting of The American Society of Heating and Ventilating Engineers, Muskoka, Ontario, Canada, June 1950.

stream, and mounted opposite thereto is an independent thermocouple for measurement of air temperature. The heated tip is supported on the four lead wires in order to favor maximum convective heat loss and to avoid appreciable disturbance of the fluid stream.

The chosen method of operation is to maintain a constant rate of heat generation in the tip and to relate the differential thermocouple electromotive force e to the velocity. (By using wire with a negligibly small change of resistance with temperature, a constant heating current will establish a constant I^2R heat generation rate for the intended range of use. A practical caution, however, is that the resistance element should be adequately aged by prolonged heating before using the anemometer, otherwise the resistance R may drift during initial use.) Advantages of the method are: (1) ease of use with several heating elements connected in series with the same constant current under remote control; and (2) convenience of remote emf measurements with speedy automatic potentiometers.

The physical principle of operation is based on a heat balance on the heated tip. Heat generation rate is equated to the convection, radiation and conduction heat loss rates:

$$q_{gen} = q_{conv} + q_{cond} + q_{rad} = \text{constant} \quad \ldots\ldots\ldots \quad (1)$$

The rate of heat conduction from the tip into the base along the lead wires, q_{cond}, can be made entirely negligible by proper design. The method used here is to treat the wires as long thin fins[6] and make them long enough to lose by convection and radiation substantially all of the heat which they conduct away from the tip.

Radiation loss to the surroundings, q_{rad}, can be minimized by: (1) taking advantage of the natural bright metallic finish of the tip and wires; (2) keeping the surface area of the heated portions small; and (3) avoiding large temperature differences to surrounding objects and calibrating in an environment similar to that intended for use. Although conditions can be visualized where q_{rad} would not be negligible, the simple theory of this paper will postulate negligible radiation.

The convection term, q_{conv}, is thus the only heat loss remaining in Equation 1 for the case being considered. The object losing heat is the tip with attached *fins* in the form of the lead wires. (The two copper lead wires for the heating element contain a small I^2R heat generation themselves, but this superposition does not affect the simplified theory.) Since the average unit convective conductance for the tip and the wires separately would vary in substantially the same manner with stream velocity[7], a single velocity factor should serve to express q_{conv}. The following limitations are necessary in order to apply the simplified theory: (1) anemometers are to be used in air only; (2) moderate tip-to-air temperature differentials are to be observed so as to minimize free convection influences; (3) anemometers are to be individually calibrated so that dimensional factors will be included in the calibration results; and (4) linear relation is postulated between differential thermocouple voltage and tip-to-air temperature difference.

The simple equation which follows from the given conditions (see Appendix for derivation) is:

$$\left(V\rho/\rho_o \right) = K \left(\frac{1}{e(C/C_o)} \right)^n \quad \ldots\ldots\ldots\ldots \quad (2)$$

where

V = stream velocity, feet per minute.
ρ = stream density, pounds per cubic foot.
ρ_0 = standard air density of 0.075 pounds per cubic foot.
e = differential tip-to-air thermocouple reading, millivolts.
n = an exponent to be established by experiment.
K = an instrument constant to be established by experiment but which can be varied for any given anemometer by changing the heating current.
C/C_0 = a correction factor for the very slight influence of air temperature and humidity, as explained in the Appendix.

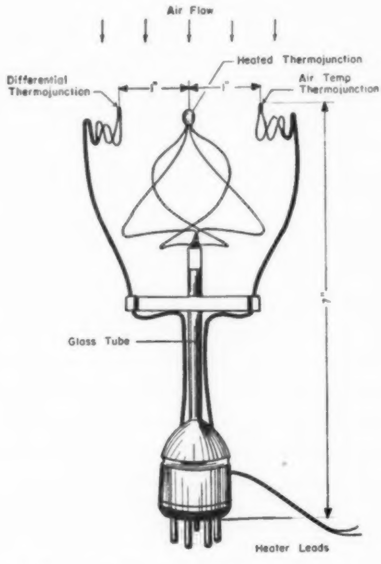


FIG. 1. HEATED THERMOCOUPLE ANEMOMETER

It should be noted that changes of K to accommodate different velocity ranges allow the anemometer differential voltage (emf) to be altered conveniently. The theory as given requires the stream temperature to be the same approaching both the heated tip and the unheated junction; if this is not the case, the prevailing stream temperature profile must be established and the observed value of e corrected accordingly.

The factor C/C_0 is nearly unity and may be neglected in many practical cases.

The anemometer form shown in Fig. 1 is intended for use in rather large spaces and with directed air movement. Variations in form are possible to suit other applications.

ANEMOMETER CONSTRUCTION

Present construction will be outlined, but future improvements due to experience in use are anticipated.

1. *Base:* A cord-end plug which snaps into a radio socket and makes thermocouple connections; a glass tube with metallic cross-arm to support the thermo-junction wires; the assembly molded together with hard-setting cement. Heater leads go out through the side of the base to separate binding posts.
2. *Thermocouples:* Made of copper-constantan duplex-insulated wire of 24 gage. Baked-on varnish over insulation will provide excellent stiffness for self support.

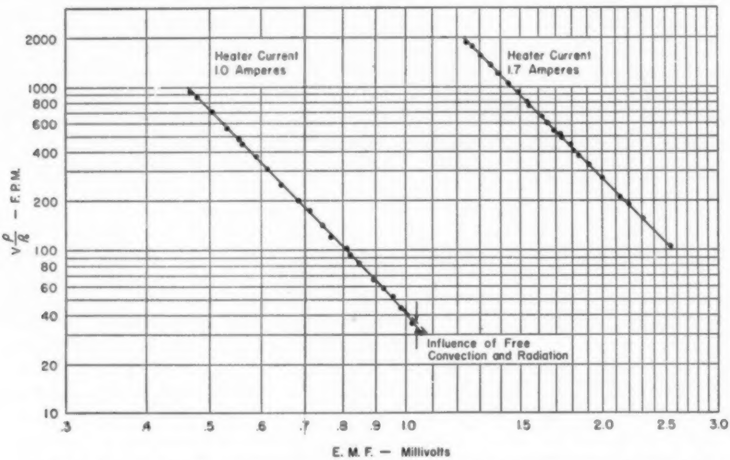
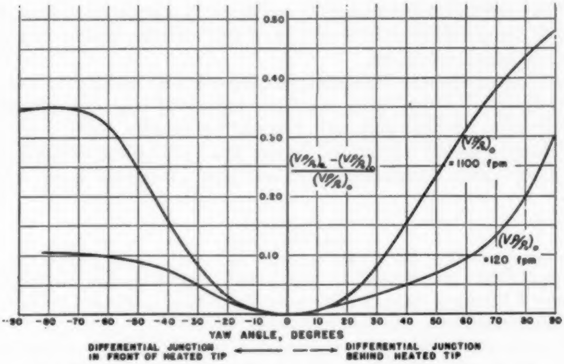


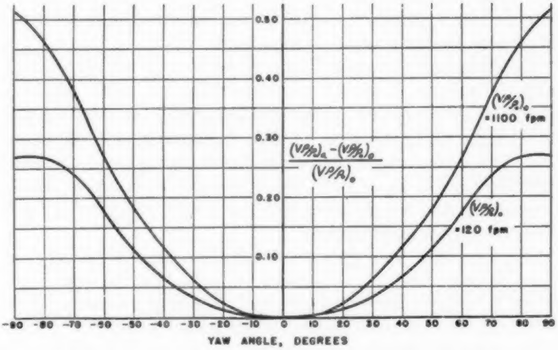
FIG. 2. SAMPLE ANEMOMETER CALIBRATION CURVES FOR $C/C_0 = 1$

3. *Heating Circuit:* Lead wires to tip are of 24-gage copper. Heating element is made of 30-gage insulated (varnish-impregnated) constantan wire wound about the thermo-junction.
4. *Heated Tip:* Four bare 24-gage wires entering tip are positioned and insulated by use of a tiny cylindrical piece of micarta through which four No. 75 holes are drilled lengthwise and which is strengthened by a few turns of reinforcing wire wound around the outside. Electrical insulation between the thermocouple and heating circuits at the thermo-junction is obtained by two coats of carefully applied and baked-on water glass. The ovate shaped metal sheath for tip is made of solder which is melted and then cast about the entire tip assembly in a metallic mold. Lead wires to tip are formed into the spiral cones and cemented in place in the end of the glass support tube.

The construction described has proved to be economical and practicable for the purpose intended. The heating current and heater resistance are chosen to give some desired limiting temperature differential with the anemometer placed tip downward in a large quieting box to obtain natural convection. (The unit of Fig. 1 happened to have a heater resistance of 0.31 ohms, and the maximum current used was 1.7 amperes.)



(A) YAW IN PLANE DEFINED BY THE STEM AND THE LINE OF THE THREE JUNCTIONS



(B) YAW IN PLANE PERPENDICULAR TO (A)

FIG. 3. YAW CHARACTERISTICS OF HEATED THERMOCOUPLE ANEMOMETER

Many variations are possible. Use of finer wires and a very small tip will constitute a sound improvement by aiding convection, although with fine wires an auxiliary support may become necessary; also smaller tips require more skill to build and have a lessened thermal capacitance.

The present design is intended for use in free-space regions, where the velocity variation across the distance of an inch or two is negligible. More compact assemblies could be built for use in regions of noticeable velocity gradients and near surfaces.

The present design is also intended for use in head-on flow, or at least in flow which is within roughly ±30 deg of being head-on. Yaw-angle calibrations have been made. Variations in designs are possible to produce an instrument which will be still more non-directional over a quite large range of yaw angles.

Use of other thermocouple materials would yield a greater electromotive force per degree of temperature differential, and this in turn would favor operation at low rates of heat generation.

The present anemometers were designed for use with automatic potentiometers indicating millivolts. With more sensitive means of emf measurement, operation at lower rates of heat generation would be aided.

Among other possible variations is the use of resistance-thermometer elements in place of a thermocouple. This has, in effect, been employed by Schmidt.[8]

Calibration

Lacking a reliable standard of comparison for velocity the anemometers have been calibrated through exposure to a known velocity under conditions representative of intended use, namely, in a free jet.

For velocities above 300 fpm a standard Pitot tube and a sensitive micromanometer were employed to establish the known reference velocity. For lower velocities a rather lengthy fit-and-try method was necessary; in this, velocity profiles were taken with an anemometer traversing across the jet outlet and the calibration established so that the mean of the velocity profiles agreed with flow rate measured by a standard orifice in the jet air supply system. Thorough double checking was the rule in all of this work.

The type of calibration obtained is shown in Fig. 2, wherein C/C_o has been taken as unity. Equation 2 is substantiated thereby, with the lower heating current favored for the very low velocities. Measurements below about 30 fpm are below the range of application of this instrument. A slight influence of free convection and radiation is noticeable below 50 fpm, and while this could have been removed by using a still lower heating current, such a move was not deemed necessary and would have had the disadvantage of further lowering the emf at the automatic potentiometer.

Data on the yaw-angle response are shown in Fig. 3. $V\ \rho/\rho_o$ denotes the velocity reading for a given yaw angle and $(V\ \rho/\rho_o)_o$ is the reading for head-on flow. Accepting a ± 5 percent reading shift as being tolerable, it follows that the anemometers may be out of alignment with the air stream by ± 25 deg to ± 30 deg with no serious consequences, depending upon the velocity being measured. The data also illustrate the influence of the thermo-junctions in shielding the air flow from the heated tip at the greater yaw angles.

Calibration data were checked with air temperatures varying from 40 to 120 F and for humidity ratios from 0.002 to 0.016 (lb vapor) per (lb dry air). The density factor ρ/ρ_o and the correction factor C/C_o were found sufficient to yield a single curve for the range of air states encountered.

Recordings of the thermocouple differential voltage in the far regions of a jet have indicated that the anemometer responds adequately to low-frequency low-velocity pulsations, such that both time-mean data and a measure of the pulsation range may be obtained.

Conclusions

As a result of the investigation it may be concluded that:

1. The heated-thermocouple anemometer, combined with an air-temperature thermocouple, is a simple and useful tool for room air distribution research.

2. Many design variations are possible beyond the simple unit described herein. These offer promise for diverse uses in air distribution problems.

ACKNOWLEDGMENT

Full credit is due Prof. L. G. Seigel of Case Institute of Technology for having given freely of his earlier experience with heated-thermocouple anemometers when the current A.S.H.V.E. project was being planned.

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APPENDIX

DERIVATION OF EQUATION 2 FOR ANEMOMETER PERFORMANCE

Adopting the simplifications discussed in the paper Equation 1 becomes

$$(h_t A_t + h_w A_w \eta)(t_t - t_o) = q = \text{a constant} \quad \ldots\ldots \quad \text{(A-1)}$$

where

h_t = mean unit convective conductance for the tip, Btu per (hour) (square foot) (Fahrenheit degree).
h_w = mean unit convective conductance for the lead wires, Btu per (hour) (square foot) (Fahrenheit degree).
A_t = tip surface area, square feet.
A_w = wire surface area, square feet.
η = fin effectiveness[6] of the lead wires, dimensionless.
t_t = tip temperature, Fahrenheit degrees.
t_o = stream temperature, Fahrenheit degrees.
q = electrical heat generation rate, Btu per (hour).

For the application intended, the unit convective conductances are postulated to be functions of the physical properties of the air and the respective Reynolds numbers. In these Reynolds numbers the geometrical factors are constant for any one anemometer. This leaves the conductances dependent upon the mass velocity of the

air stream and the two physical properties, viscosity and unit heat capacity. Taking ρ_o as a fixed standard density, the mass velocity may be expressed as $V\ \rho/\rho_o$.

The fin effectiveness of the wires varies somewhat with the conductance h_w or, in turn, primarily with $V\ \rho/\rho_o$. While a constant ratio of h_t to $h_w\eta$ would not be strictly expected for all velocities, the simplest supposition would be that such a constant ratio prevailed; let this be postulated subject to experimental validation, and call the ratio C_1. Departures from this assumption would be expected to be most noticeable at low velocities. The common mode of expressing the convective conductance in terms of the Reynolds number (over a limited range) indicates an expression of the form

$$\frac{h_t}{c_p\left(\frac{V\rho}{\rho_o}\right)} = C_2\left(\frac{V\rho}{\mu\rho_o}\right)^a \quad . . . . \text{(A-2)}$$

where

C_2 = an empirical dimensional constant.
a = an empirical constant, dimensionless.
c_p = unit heat capacity of moist air, Btu per (pound) (Fahrenheit degree).
μ = absolute viscosity of moist air, (pound) (second) per (square foot).

A further assumption is that within the application intended the slope of the curve relating temperature difference and thermocouple electromotive force is nearly constant and therefore may be taken as the average derivative,

$$\left(\frac{dt}{de}\right)_{avg}$$

where the subscript denotes the slope at the average of the air temperature (sensed by the unheated anemometer junction) and the heated-tip temperature. This slope may be calculated as a function of average temperature for the particular thermocouple wire involved.

With the substitutions given, Equation A-1 becomes

$$C_2\left(\frac{V\rho}{\mu\rho_o}\right)^a\left(c_p\,\frac{V\rho}{\rho_o}\right)(A_t + C_1A_w)\left(\frac{dt}{de}\right)_{avg} e = q = \text{a constant} \quad . \text{(A-3)}$$

Combining all constants, including the surface areas as constant factors for any one anemometer, a convenient calibration relationship follows from a plotting of

$$\left(\frac{V\rho}{\rho_o}\right) \textit{vs.}\ e(C/C_o)$$

where

$$C = c_p\mu^{-a}\left(\frac{dt}{de}\right)_{avg} \quad . . . . \text{(A-4)}$$

The factor C is evaluated with the air properties taken at the average of the unheated-tip (or air) and heated-tip anemometer temperatures and for the prevailing humidity ratio.

The factor C_o is taken as a *reference value of* C for some temperature and humidity ratio in the middle of the intended range. This makes (C/C_o) nearly unity, such that it may be neglected if desired.

The exponent a may be determined from a log-log plotting of the relationship

$$(V\rho/\rho_o) = \frac{\text{constant}}{(e\ C)^{\frac{1}{a+1}}} \quad . . . . \text{(A-5)}$$

In practical use a family of curves giving C/C_o as a function of the average of the heated and unheated-tip temperatures and the humidity ratio will expedite the reduction of data.

For the anemometers used at the A.S.H.V.E. Research Laboratory the magnitude of C/C_o is within the maximum limits of 0.978 and 1.013 over the range of air states noted. This factor, however, brings all calibration and calibration-check velocity data together within a probable tolerance of approximately ± 5 percent. Note, however, that this tolerance applies to ideal or calibration conditions and need not be representative of all conditions of use.

DISCUSSION

Elliot Godes, New York (Written): The author is to be congratulated upon his interesting contribution. Of particular interest and value is the fact that errors due to ambient temperature changes were evidently reduced to a minimum.

About four years ago the company with which I am associated developed, built and tested two instruments based upon the same general principles, but different in construction. One of these instruments had five, the other had 12, sensitive elements. The results obtained, however, were not entirely satisfactory for the following reasons:

1. Difficulty in obtaining uniformity of multiple heated thermocouples.
2. Excessive directional effect of air flow on velocity readings.
3. Excessive effect of ambient temperature on readings.
4. Slow response.

Far more useful results were achieved by development of instrumentation based upon the principle of the heated resistance thermometer, although the main reason for switching to this principle was the desire to obtain a direct reading instrument with higher temperature accuracy.

In the heated-thermocouple anemometer described, a way has evidently been found to minimize the ambient temperature effect and to obtain an adequately speedy response at the expense of a decided reduction of sensitive element mass and structural sturdiness. From Fig. 1 it is difficult to believe that the unit could stand the impact of velocities up to 2000 fpm, head-on or otherwise. Probably it would be advisable to use more conventional instruments for velocities of these magnitudes, particularly if they are as clearly directional as those measured by this anemometer.

Another point to be considered is the difficulty in reading the very small variations in electromotive force caused by considerable differences in air velocities. Thus, a variation of 0.1 millivolt may correspond to as much as the difference between 430 and 1000 fpm for a 1 ampere heater current or the difference between approximately 720 and 1000 fpm for a 1.7 ampere heater current. It is, therefore, necessary to obtain readings with an accuracy of at least 0.01 millivolt. The original reading shift is said to be within ± 5 percent and any errors in converting millivolt readings into velocities would have to be added to the error of the instrument.

In view of the required elaborate measuring equipment and the time consumed for taking indirect readings, this degree of assured accuracy must be considered to be low.

It is, therefore, hoped that the A.S.H.V.E. Research Laboratory will continue this development work and particularly that it will also investigate the possibilities of an instrument having a still lower effect of air flow direction on the reading (so as to permit determination of turbulent air motion) and capable of handling multiple sensitive elements without sacrificing any of the desirable features of the present anemometer.

Raymond Mancha, Pittsburgh: How does one determine air flow direction; also, is it possible that the geometry of the instrument can be modified for probing?

LINN HELANDER, Manhattan, Kans.: In our research project on the *Downward Projection of Heated Air,* the flow and temperature at any given point fluctuate simultaneously. Because of this, the commercially available heated wire anemometers that we have tried have not proved satisfactory.

Will the heated thermocouple anemometer described in the paper give a satisfactory indication of the velocity of a stream when the velocity and temperature of the stream fluctuate simultaneously?

D. D. WILE, Los Angeles: Mr. Nottage's paper is particularly valuable as an analysis of the various factors that influence the operation of this type of instrument. As stated by him the design is subject to wide modification.

Some years ago I constructed several types of heated thermocouple anemometers for measuring air motion over thermostats used to control room temperature. One of these anemometers used in conjunction with a portable microammeter was operated from two small flashlight batteries and had a measuring tip no longer in diameter than an ordinary match. This instrument was very successful in measuring air currents from approximately 20 up to 500 fpm.

AUTHOR'S CLOSURE: With thanks to the discussers, it is pertinent to point out that the paper was intended to bring out physical principles and not to *solve* the anemometer problem.

In reply to Mr. Godes, the unit described was developed for a particular laboratory use and does not compete with anemometers intended for field service. Accessory equipment is not elaborate but it is not readily portable. Adoption of a thermocouple was dictated by the availability of automatic recording potentiometers in the stock of laboratory instruments. Moreover, these potentiometers forced a compromise by requiring millivolts, although the design analysis indicated the superiority of operating in the *microvolt* range. Use of thermocouple materials having a higher *emf* per unit of temperature difference than copper and constantan also was desired from design calculations, but again a compromise was made. Performance has been satisfactory despite these compromises.

Recent marketing of small, highly compact and standardized resistance-thermometer elements, in combination with automatic recording instruments, is worthy of note by interested persons. The unit described actually was designed for the range 30 to 800 fpm. The high range has not been used and was noted primarily as demonstration of principle. The change in *emf* per unit change in velocity may be controlled in design. Basic dependence upon convective heat transfer, combined with recognition of the billowing, pulsing and wandering nature of free air movement, indicated a practical limit to desirable response sensitivity. It is not claimed, however, that the design described has reached this limit.

The unit has been used up to 3000 fpm. The structure is more rugged than might appear. Refinements of accuracy are attainable, but then the device would pass out of the *simple* classification. Directional response is controllable in design. A non-directional instrument was not sought. For the application being made, velocity is considered to be a vector quantity, and direction is measured by other means which remain for future reporting.

In reply to Mr. Helander, for a true study of rapid fluctuations it is necessary to consider elaborate and expensive instrumentation, as represented by the hot-wire anemometers which have been highly developed in aerodynamic laboratories. For low-frequency, low-amplitude pulses, the effects of temperature are contained in the calibration relationship giving $(V\rho/\rho_0)$ *vs.* (eC/C_0). For use in traversing non-isothermal flow, simultaneous records of the differential *emf* and the air temperature are planned. This is one reason why the instrument carries its own air-temperature thermocouple.

In reply to Mr. Mancha, the unit described is not, in itself, suitable as a direction indicator. Mr. Wile has nicely emphasized the appealing flexibility of the basic principles.

1403

No. 1403

EFFECT OF TEMPERATURE ON BALANCE OF FORCED WARM-AIR SYSTEMS

By N. A. Buckley*, S. Konzo**, J. M. David†, and T. L. Towne††,
Urbana, Ill.

A forced warm-air heating system which maintains desired indoor temperatures in all occupied spaces, and under all load conditions, is considered to be in balance. Normally any unbalance which occurs in a system can be traced directly to unusual weather conditions such as high velocity winds, or intense solar radiation. However, some cases have been reported in which the unbalance that was observed could not be attributed to unusual weather conditions alone.

Up to this time no satisfactory explanation has been offered for this phenomenon although the evidence indicates that the unbalance was in some ways related to the total volume of air flowing in the system and the relative elevations of the supply registers. In some extreme cases it has been observed that a warm-air register located in the basement may become sufficiently unbalanced so that under mild weather conditions the supply of warm air is reduced to a negligible quantity. In view of the widely varying explanations as to the cause and extent of this unbalance, a series of laboratory tests under controlled conditions was conducted. This paper gives a condensed summary of the investigations and offers an explanation of the factors which affect the balance of an air distribution system. Furthermore, a method is presented for predetermining the location and magnitude of unbalance for any given system.

Description of Test Apparatus

The test arrangement, shown schematically in Fig. 1, consisted of a four-stack system, in which register openings were located on three different levels above the furnace. This arrangement simulated a heating installation for a two-story residence in which a basement room is heated. The basement outlet may also be considered to be approximately of the same elevation as a floor

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Presented at the Semi-Annual Meeting of The American Society of Heating and Ventilating Engineers, Muskoka, Ontario, Canada, June 1950.

register or baseboard register on the first floor. The trunk duct was of the extended plenum type[1], and its dimensions were 12 in. × 8 in. from the furnace plenum to the far end. The trunk size of this extended plenum corresponds to the trunk size at the furnace plenum for a system[2] designed in accordance with Manual 7 of the *National Warm Air Heating and Air Conditioning Association.*

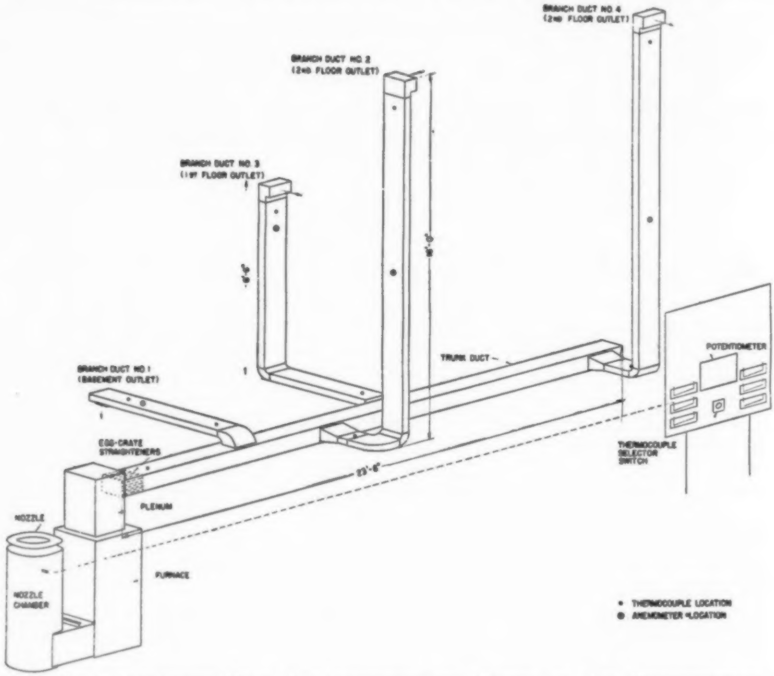

FIG. 1. LABORATORY TEST ARRANGEMENT FOR STUDY OF AIR DISTRIBUTION

Each of the four branch ducts of the system consisted of a 3¼ × 12 in. stack, but with registers omitted at the stack-head.

The blower installed in the furnace supplied the motive head necessary to overcome the various resistances of the system and to draw the desired volume of air through the *National Bureau of Standards* type nozzle installed at the return-air inlet of the furnace. The desired flow rate of air into the system was obtained by selection of the proper sized nozzle, and by the adjustment of a slide damper installed between the furnace and the nozzle chamber. Egg-crate straighteners were inserted in the trunk duct for the purpose of eliminating any swirling motion from the air entering the trunk duct or test section.

The volume flow rates out of each of the individual branches were obtained by means of a vane anemometer installed in each branch and separately cali-

[1] Exponent numerals refer to References at end of Appendix.

brated in place against the *National Bureau of Standards* type nozzles. The calibration of these anemometers was established both with recirculated laboratory air and heated air.

The temperature of the air flowing through the system was measured by means of thermocouples placed in the nozzle chamber, in the trunk duct near the furnace plenum, at each branch take-off and at each stack head. Temperatures at any six stations could be recorded continuously by means of a recording potentiometer. Periodic measurements of temperatures at all thermocouple stations were made during actual test runs with a precision type potentiometer.

For a second test arrangement, which simulated a system having register openings on only two elevations, branch duct No. 1 to the basement was closed and sealed. The instrumentation for the tests on this three-branch system was the same as that for the four-branch system.

METHOD AND PROCEDURE FOR CONDUCTING TESTS

Tests were conducted on both the three- and four-branch systems with zero heat input, as well as with heat inputs of 5000 Btu per hr, 15,000 Btu per hr, 25,000 Btu per hr, and 35,000 Btu per hr. With zero heat input to the system, the total air volume was varied over a range from about 100 cfm to 500 cfm. For each adjustment of total air volume, the flow through each branch duct was recorded and plotted as shown in Fig. 2. Similar tests were conducted for each of the four heat inputs to the furnace, and temperature measurements were made at all thermocouple stations.

RESULTS OF TESTS WITH ZERO HEAT INPUT

For zero heat input to the system, the values for branch air volumes were plotted as shown in Fig. 2, against the total air volumes. A straight line could be drawn through the points representing the data for each of the branches.

Since no density changes were involved from the inlet to the registers, the total air volume passing through the nozzle should be equal to the summation of the air volumes flowing through the stacks. An agreement within about 3 percent was obtained which was considered to be within the range of experimental error.

From the data shown in Fig. 2 it would appear that the frictional resistances, and therefore the equivalent duct lengths, of branches Nos. 1 and 2 were greater than those for branches Nos. 3 and 4. Calculated values of the equivalent duct lengths made from the best available data were: 144, 98, 73, and 67 ft respectively for branches Nos. 1, 2, 3, and 4. These calculated values of equivalent length, therefore, substantiated the results obtained from the test data.

For all practical purposes, the four curves shown in Fig. 2 converge toward the origin. The slight discrepancy of the data for branch No. 3 in this respect has been attributed to the uncertainties in measuring the low air volumes. In the case of zero heat input the evidence indicates that as the total air volume was changed, the flow through each of the branch outlets was proportionally changed. Hence, it may be concluded that no unbalancing action occurred in this four-stack arrangement. Similar results were obtained with the three-stack arrangement and with zero heat input.

RESULTS OF TESTS WITH HEAT INPUT

When heat was added to the duct system, proportional air delivery to each of the four branches was no longer obtained. For low total volumes, flow through one of the branch ducts decreased to zero with a resulting increase in flow in the three remaining branches. This is illustrated by the second group of curves from the top of Fig. 3 which shows complete data for an actual test in which the heat input was 35,000 Btu per hr. For example, branch duct No. 1 delivered 92 cfm, or 18.4 percent when the total volume was 500 cfm, and zero

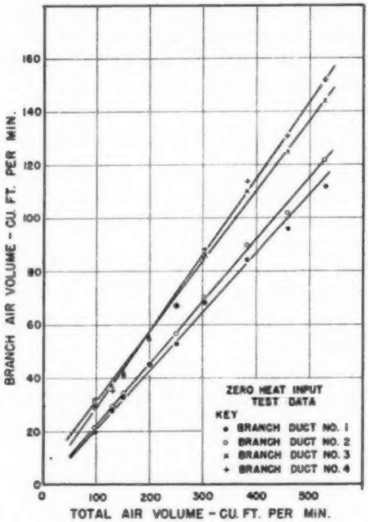

FIG. 2. DISTRIBUTION OF AIR OBTAINED WITH ZERO HEAT INPUT

cfm when the total volume was about 240 cfm. It was also observed that for total volume less than 240 cfm, the direction of flow in branch duct No. 1 was reversed and cool laboratory air was introduced into the trunk duct. As may be noted from Fig. 1 branch duct No. 1 led to a basement outlet and was at the same elevation as the trunk duct. In all cases the observed air volumes for the branch ducts were corrected to laboratory conditions of temperature and pressure, before plotting.

Air volume alone is not a criterion of heat delivery to a given space, since air temperature is also involved. For the purpose of determining the register delivery, air temperature measurements were made, and the values of temperature rise were plotted as shown in the top group of curves in Fig. 3. Since the tests were made with a constant heat input, any decrease in total air volume should result in an increase in temperature rise. This was observed to be the case for total air volumes greater than about 325 cfm. For total volumes less than 325 cfm the temperature rise for branch duct No. 1 decreased sharply as

the result of excessive heat loss[3] for the duct accompanying the low velocities in the duct.

The third group of curves from the top of Fig. 3 represent the register deliveries which were defined as:

$$H = 60 \cdot C_p \cdot Q \cdot \rho \cdot \Delta t,$$

where

H = register delivery, Btu per hour.
C_p = specific heat of air, Btu per (pound) (Fahrenheit degree).
Q = branch volume flow rate, cfm.
ρ = air density, pounds per cubic foot.
Δt = temperature rise inlet to register, Fahrenheit degrees.

From the standpoint of distribution balance, the percent of total register delivery is of more significance than the actual magnitudes of register delivery. These percentages were plotted as shown in the bottom set of curves in Fig. 3. A constant percentage over a wide range of total volumes, as shown for branch duct No. 3, indicates that no unbalancing action occurred in this particular branch. The unbalancing action is clearly shown for branches Nos. 1, 2, and 4. In the case of branch No. 1, the register delivery decreased to zero as the total volume was decreased, whereas for branch ducts Nos. 2 and 4 the converse was true. The data shown in Fig. 3 apply only to a fixed heat input of 35,000 Btu per hr. It was necessary, therefore, to investigate the conditions existing for heat inputs between zero and 35,000 Btu per hr, and the results of such tests are shown in Figs. 4 and 5.

Up to this point in the discussion, Fig. 2 indicated that no unbalancing action occurred with zero heat input, whereas Fig. 3 indicated considerable unbalancing action with 35,000 Btu per hr heat input. The values for branch air volumes shown in Fig. 4 for heat inputs of 5,000 Btu per hr, 15,000 Btu per hr, and 25,000 Btu per hr indicate that the unbalancing action is a function of the heat input. That is, the curves for the 5,000 Btu per hr input more nearly approximated those shown in Fig. 2 for zero heat input, while the curves for 25,000 Btu per hr approached those shown in Fig. 3 for 35,000 Btu per hr. In all cases the unbalancing action was more pronounced for low total volumes flowing. Furthermore, branch duct No. 1 was always the branch which reached zero air delivery or the drop-out point.

Curves showing the percent of total register delivery for the four heat inputs are given in Fig. 5 and confirm the evidence shown in Fig. 4. Branch duct No. 3 shows relatively constant percentage of delivery over a wide range of total air volumes and heat inputs. Branch ducts Nos. 2 and 4 which were second story stacks showed a tendency to deliver a higher percentage of total register delivery with lower total air volumes. On the other hand, branch duct No. 1, which served as a basement outlet, reached a drop-out point.

Curves similar to those shown in Fig. 5 were obtained for the three-stack arrangement consisting of branch ducts Nos. 2, 3, and 4, and are given in Fig. 6. In this case where the basement outlet was not involved, the stack which reached the drop-out point was branch No. 3 which served a first story register. In this case branch duct No. 4 showed a relatively constant percentage of total register delivery, and stack No. 2 showed an increasing percentage with lower total air volumes. From the evidence shown in Figs. 5 and 6, it may be noted that the registers at the lowest elevation in a given duct system reached a drop-out point.

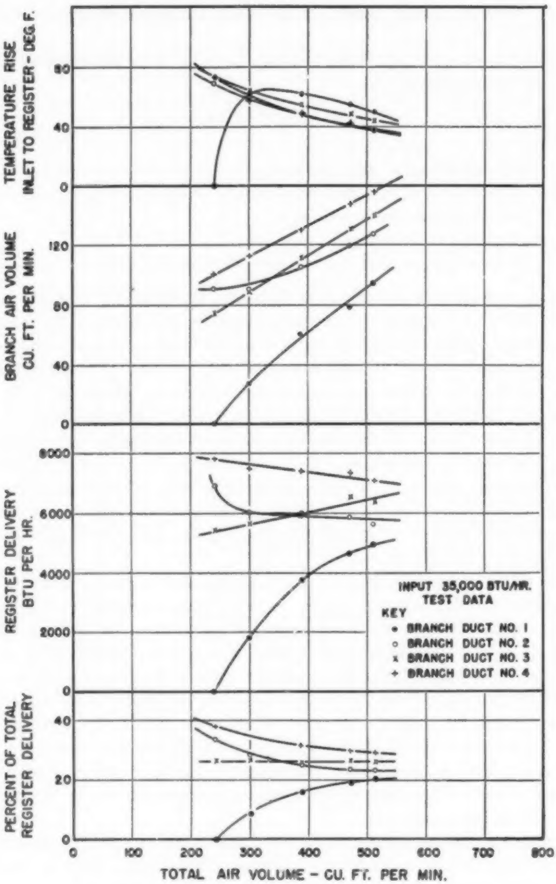

FIG. 3. TEST RESULTS OBTAINED WITH 35,000 BTU PER HR

FACTORS WHICH CAUSE UNBALANCE

Any explanation of the unbalancing action must account for the phenomena shown in Figs. 5 and 6. It should be noted that no change was made in the duct system and that no appreciable changes in external temperature or pressure were observed. The only difference consisted of changes in the properties of the fluid flowing. In the case of the data shown in Figs. 2 and 3, the changes in density of the air were due to the addition of heat. This would lead to the conclusion that the unbalancing action was related to stack effect.

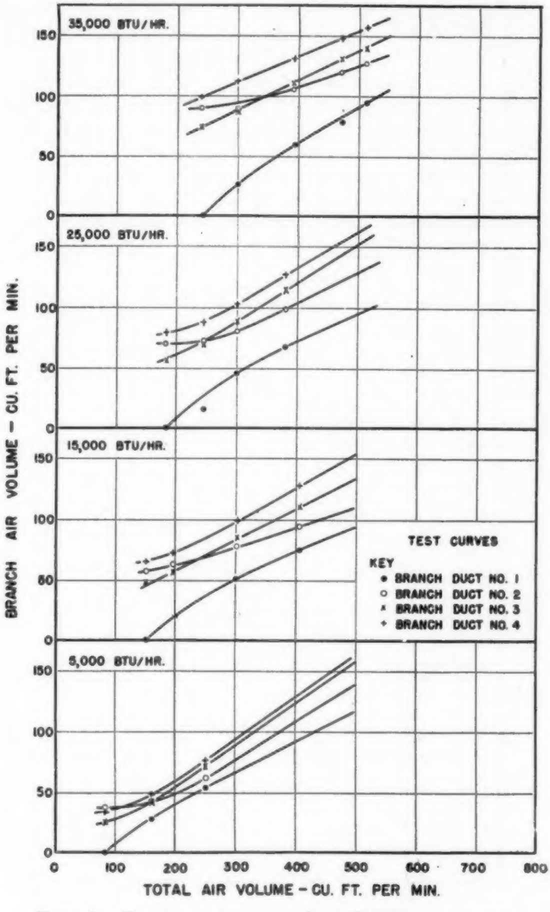

FIG. 4. DISTRIBUTION OF AIR OBTAINED WITH THE FOUR-STACK SYSTEM

For the gravity warm-air heating system, the expansion of warm air and the consequent reduction in density furnishes the motive head necessary to cause air circulation. In a forced-air system a similar motive head is present. This motive head, or stack effect, influences the circulation through the system to a greater or lesser degree depending upon the temperature of the flowing air and the pressures created by the furnace blower. In other words, the *total* motive head acting to produce flow of air through a given branch duct is composed of a stack effect and a pressure head produced by the furnace blower.

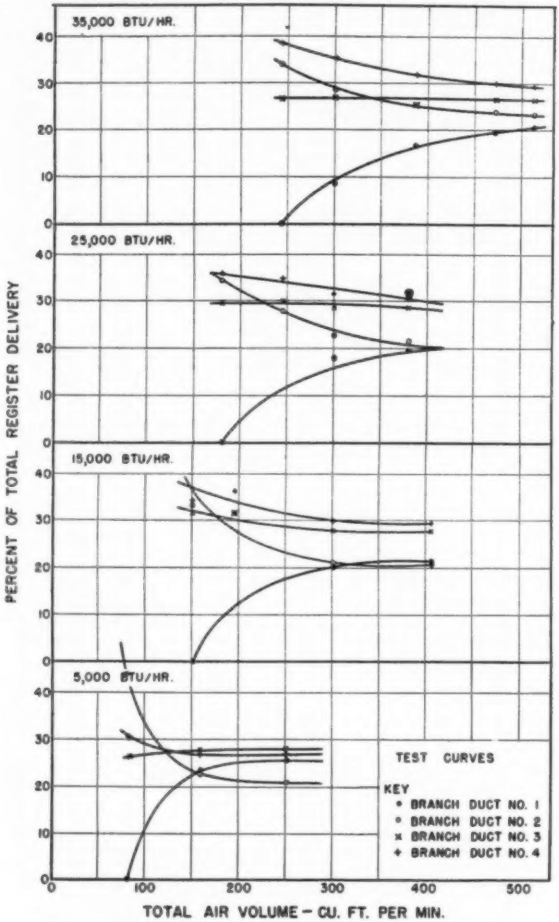

FIG. 5. PERCENT OF TOTAL REGISTER DELIVERY FOR THE FOUR-STACK SYSTEM

Where differences in register elevations exist, the stack effects in the various branches are not the same since both air temperatures and stack heights are involved.

If the total motive head in some of the stacks of a forced-air system were disproportionately great, due to presence of stack effect, these stacks would tend to take more than their share of the total air volume handled, thus robbing the stacks having less total motive head. For the test arrangement shown in Fig. 1,

branch ducts Nos. 2 and 4 have stack heights of 16 ft above the trunk duct, whereas branch duct No. 1 has zero stack height. It would be expected that branch ducts Nos. 2 and 4 would tend to rob branch duct No. 1, if the stack effect is of sizable magnitude relative to the pressure effect of the furnace blower. The greatest unbalance would be obtained when the temperature of the air flowing was high (large stack effect) and the total air volume was low (small effect of blower pressure). It is conceivable that the total motive head of the lowest stack could be zero when the stack effect of the higher stacks equals the blower pressure available at the take-off of the lower stack. For these conditions the flow in the lower stack would become zero. This explanation is therefore offered for the unbalancing observed in Figs. 5 and 6.

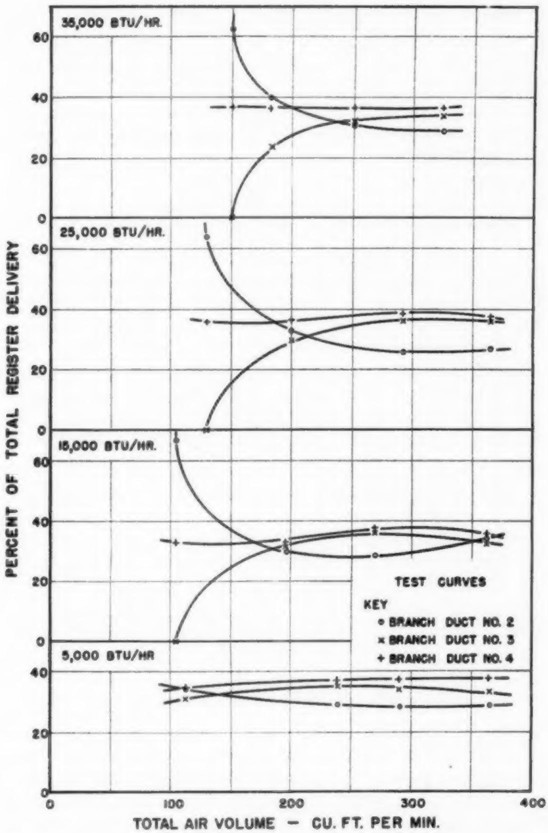

FIG. 6. PERCENT OF TOTAL REGISTER DELIVERY FOR THE THREE-STACK SYSTEM

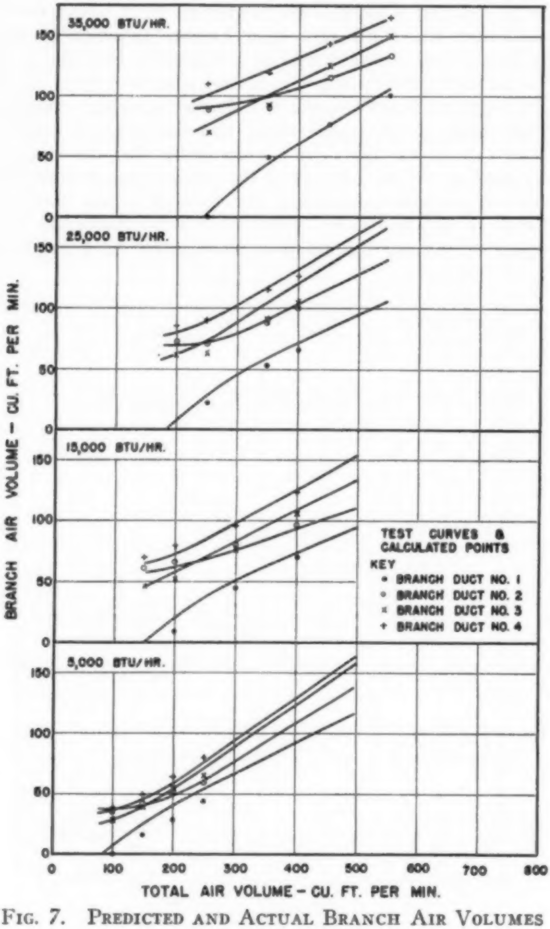

FIG. 7. PREDICTED AND ACTUAL BRANCH AIR VOLUMES FOR THE FOUR-STACK SYSTEM

RESULTS OF PREDICTED UNBALANCE

If the foregoing explanation is adequate it should be possible to make a prediction about when and to what extent unbalance will occur. For example, in a given system with a known quantity of air flowing and a known heat input, it should be possible to calculate the system resistance to flow, and the stack effect. Such predictions were made, as discussed in detail in the following Appendix, and the results are shown as plotted points in Fig. 7. The solid curves shown in the same figure represent actual test data, and are the same as those

shown in Fig. 5. Excellent agreement was obtained between the predicted and actual test data. Thus the explanation which has been presented adequately accounts for the unbalancing action shown in Figs. 5 and 6.

APPLICATION OF RESULTS

Specifically the preceding discussion applies only to the extended plenum duct system shown in Fig. 1. However, since the actual test results and the predicted results were in excellent agreement, it would appear that this method of prediction could be applied to duct systems other than those tested in the laboratory.

Previously it was noted that the unbalancing action was a function of the heat input and was more pronounced for low total volume flowing, and that in the four-stack system, branch duct No. 1 was always the branch which reached the *drop-out point*. Values of total air volume at the *drop-out point* for various heat inputs were obtained from Figs. 4 and 5 and were plotted against the corresponding heat input as is shown in Fig. 8. The solid line curve drawn through the points represents the total air volume for which branch duct No. 1 gives zero air delivery for a range of heat inputs. A second curve was established in Fig. 8 for that value of the total air volume at which unbalance became readily apparent. For example, from Fig. 5 for a heat input of 15,000 Btu per hr the deviation of the curves from the horizontal line, representing a constant percentage of total register delivery, became apparent at approximately 260 cfm. This volume together with similar values for other heat inputs is represented by the broken-line curve in Fig. 8 and has been designated as the *break-away* curve. The area between the curves for *drop-out* and *break away* is indicated as the *critical zone*. Any method of control or operation which will allow the heating system to operate in this *critical zone* will result in unbalanced heat distribution.

The question arises as to how a heating system with a single rate of input can operate at any other input. The methods of control which are commonly employed produce widely varying heat inputs by intermittent operation of the burner. For example, a gas furnace with an input rating of 40,000 Btu per hr, could be operated with a three minute *on* cycle and a nine minute *off* cycle during mild weather. In this case the average input will be approximately 10,000 Btu per hr.

If the total air volume were adjusted to provide a 100 deg temperature rise from inlet to bonnet, in accordance with the *continuous air circulation* principle[4], a total air volume of approximately 400 cfm would be circulated when a rated heat input of 40,000 Btu per hr was maintained. Then, during extremely severe weather when the furnace was operating continuously, at maximum capacity, the system would operate within the fringe of the critical zone as is indicated in Fig. 8.

For all weather conditions in which less than 35,000 Btu per hr was required, which would be the case for practically the entire season, the system would operate above the critical zone and no unbalancing would be experienced. Critical unbalancing should not be experienced for a duct system whose air volume is in conformity with the *continuous air circulation* principle.

Normally, a system which is adjusted to deliver a given air quantity can be expected to maintain this delivery indefinitely. However, if any change in system resistance occurs so that a much smaller volume is being circulated then

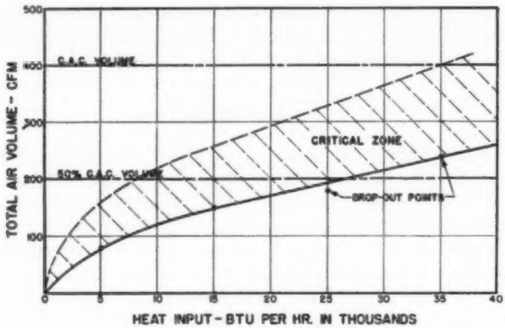

FIG. 8. APPLICATION OF TEST RESULTS TO SINGLE- AND TWO-SPEED BLOWERS

the possibilities of obtaining an unbalanced distribution increase. For example, if the air filter resistance increases over a long period of time, the total air volume will decrease. A point will eventually be reached when the total air volume will be only a fractional part of the initial air volume. For the case shown in Fig. 8 in which the system was balanced for an initial total air volume of 400 cfm, some difficulties in distribution balance can be expected when the total air volume decreases to a value of about 300 cfm or less. A similar analysis applied to the case of a two-speed blower indicates that for certain methods of operation an unbalance can be expected. For example, if the blower were adjusted so that the total air volume flowing was 200 cfm for the low speed operation in mild weather and 400 cfm for high speed operation in severe weather, then some difficulty can be expected in balancing at the low speed during mild weather. In this example during low speed operation the system would operate in the critical zone for weather conditions requiring more than about 8000 Btu per hr input. When the weather became severe and the blower operated at the higher speed the system would again operate above the critical zone. On the other hand, if the blower were adjusted so that 300 cfm was delivered at low speed and that 500 cfm was delivered at high speed, unbalance should not be experienced if the change to high speed operation was made at an input of 25,000 Btu per hr or less.

In a similar manner the operating conditions for variable speed blowers may be analyzed as illustrated in Fig. 9. The curve designated as *variable speed pulley A* illustrates one type of operation in which the speed of the blower is low for low heat inputs and increases proportionately with heat input. Maximum speed is reached when maximum heat input is obtained during severe weather conditions. For this method of operation the curve representing the volume delivery of the blower falls below the drop-out curve for heat inputs less than about 11,000 Btu per hr; and falls within the critical zone for heat inputs between 11,000 and 35,000 Btu per hr. Therefore, for this method of operation, unbalance in distribution of air quantities should occur over a wide range of weather conditions.

The curve designated as *variable speed pulley B* illustrates a more successful

type of operation of a variable-speed blower in which the blower starts operating at 50 percent of full speed and increases gradually in speed with increasing heat inputs. For this method of operation the curve representing the volume delivery of the blower lies above the critical zone over the entire range of heat inputs. In this case no unbalancing action is to be expected.

It should be emphasized that the preceding discussion relative to Figs. 8 and 9 was based upon results obtained from a test arrangement in which the registers were located at three elevations. In the three-branch system which had registers at only two elevations, the unbalancing action was found to be less severe. Since the unbalancing action has been shown to be due to the differences in stack heights of the various branches, no unbalance should be expected in a system having all registers at the same elevation.

CONCLUSIONS AND SUMMARY

For the conditions described the following conclusions could be drawn:

1. With zero heat input, any change in the total air volume delivery was accompanied by a proportional change in the flow through each of the branch outlets.

2. When heat was added to the duct system, proportional air delivery to each of the branches was no longer obtained. For the arrangement tested branch ducts Nos. 2 and 4 to second story registers showed a tendency to deliver a higher percentage of the total volume delivery and particularly with low values of the total volume delivery. On the other hand branch duct No. 1 which served a basement outlet reached a drop-out point or zero air delivery.

3. A comparison of tests in which no change was made in the duct system, and no appreciable change was observed in pressures and temperatures external to the duct system, indicated that the addition of heat to a duct system in which registers are located at more than one elevation results in an unbalancing action. This action was attributed to differences in stack height.

4. An analytical method was devised for predicting the air distribution in an extended plenum duct system with heat added to the system. In this analysis, the stack effect was superimposed upon the pressure supplied by the furnace blower. Good agreement was obtained between predicted and actual results, thus confirming the validity of the stack effect as a possible cause of unbalanced distribution.

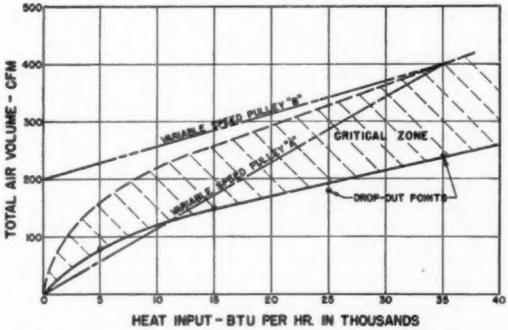

FIG. 9. APPLICATION OF TEST RESULTS TO VARIABLE SPEED BLOWERS

5. The application and extension of the test results obtained from the four-branch arrangement to the following cases were discussed in detail: *(a)* single-speed blower, *(b)* effect of filter resistance, *(c)* two-speed blowers, and *(d)* variable speed blowers. On the basis of this analysis it would appear that the unbalancing action could be avoided by proper adjustment of volume flow rates to heat input.

6. In the three-branch system which had registers at only two elevations the unbalancing action was found to be less severe than that in the four-branch system. Furthermore, no unbalance should be expected in a system having all registers at the same elevation.

ACKNOWLEDGMENT

This paper is a report of a laboratory investigation conducted under the terms of the cooperative agreement between the University of Illinois Engineering Experiment Station and the *National Warm Air Heating and Air Conditioning Association.* Acknowledgment is made to the Research Advisory Committee of the Association for advice and counsel received.

Acknowledgment is also made to F. S. Alexander, former graduate student at the University of Illinois for service rendered in construction of the test arrangement and conducting of initial tests.

APPENDIX

METHOD FOR PREDICTING AIR DISTRIBUTION IN A DUCT SYSTEM

An analytical method was used to predict the air distribution in the extended plenum duct system tested in the laboratory. The step by step procedure will be outlined with explanatory paragraphs to clarify each step. As a further illustration of the method, sample calculations were made for the laboratory arrangement and the values are tabulated in Table A. For the purposes of this analysis a number of assumptions were made and are explained in detail under the appropriate items.

The heat input to the furnace can be determined for a given furnace from the measured rate of fuel input and the heating value of the fuel. The bonnet capacity or heat output is then the product of the heat input and the bonnet efficiency. For the electric furnace used in the laboratory test arrangement, a bonnet efficiency of 100 percent was assumed. A heat input and output of 25,000 Btu per hr was used for the example shown in Table A.

EXPLANATION

In the following explanation, the paragraph numbers correspond to items in Table A.

1. *Branch duct number.* Branch duct numbers are shown in Fig. 1.

2. *Equivalent length of branches, feet.* The equivalent lengths of the branch ducts were the sum of the actual lengths of the branch ducts, plus the equivalent lengths of the fittings[1, 2].

3. *Pressure available at the branch take-off inches of water.* In normal design procedure for a forced-air system, the maximum available pressure at the discharge of a furnace is assumed to be 0.10 in. of water; this would apply only to the maximum volume delivery. For the purpose of covering a wide range of air volumes it would be possible in the analytical procedure to select a number of values of available pressure within the range from 0 to 0.10 in. of water. Actually the calculations were based upon four assumed pressures of 0.012, 0.018, 0.035, and 0.045 in. of water. In the example shown in Table A calculations are given only for one of these assumed pressures, namely 0.035 in. of water. For the arrangement shown in Fig. 1 calculations indicated that the frictional losses in the trunk were negligibly small. Hence, the pressures available at all branch take-offs were considered to be equal.

TABLE A—VALUES OBTAINED FROM SAMPLE CALCULATIONS FOR DETERMINING AIR DISTRIBUTION

(HEAT INPUT = HEAT OUTPUT = 25000 BTU PER HR)

1. Branch duct No.	1	2	3	4
2. Equivalent length of branches, ft	114	98	73	67
3. Total pressure at the branch take-off, in. of water	0.035	0.035	0.035	0.035
4. Pressure available per 100 ft of branch duct, in. of water	0.031	0.036	0.048	0.053
5. Air volume flowing, zero heat input, cfm	75	82	96	101
6. Temperature difference inlet to register, deg	49	36	41	40
7. Density of air in the stack, lb per cu ft	0.0668	0.0684	0.0678	0.0679
8. Air volume flowing, corrected for density, cfm	81.7	87.4	103	108
9. Loss per 100 ft of branch duct, from chart, in. of water	0.036	0.041	0.055	0.061
10. Loss per 100 ft of branch duct, corrected for density, in. of water	0.033	0.038	0.051	0.069
11. Pressure to deliver corrected volume, in. of water	0.037	0.037	0.037	0.037
12. Stack effect, in. of water	0	0.014	0.008	0.016
13. Combined stack pressure, in. of water	0.024	0.052	0.040	0.056
14. Stack pressure per 100 ft of branch, in. of water	0.021	0.053	0.055	0.084
15. Stack pressure per 100 ft of branch, corrected for density, in. of water	0.023	0.056	0.059	0.090
16. Volume of heated air flowing, cfm	64	104	107	135
17. Volume corrected to inlet air temperature, cfm	59	98	100	126
18. Percent of total volume	15.4	25.6	26.1	32.0
19. Branch air volumes for total volume of 350 cfm	53.8	89.6	91.4	115.2

4. *Pressure available per 100 ft of branch duct, inches of water.* For an assumed pressure in the trunk duct and for known equivalent length of branch duct the friction loss per 100 ft of branch duct can be determined by the following equation:

$$\text{(Item 4)} = \frac{\text{(Item 3)} \times 100}{\text{(Item 2)}}$$

5. *Air quantity flowing, zero heat input, cubic feet per minute.* The 3¼ in. × 12 in. branch ducts shown in Fig. 1 were considered to be equivalent to a 6.6 in. round pipe. The volume of air flowing through the duct could then be determined from the friction chart[5].

6. *Temperature difference inlet to register, Fahrenheit degrees.* For the known heat input of 25,000 Btu per hr and total air volume flowing (Item 5), the temperature rise from inlet to bonnet of the furnace could be determined. The temperature drops of the air from the bonnet to the registers could be calculated from curves shown in a previous paper[3]. The values for Item 6 could then be obtained by subtracting these values from the temperature, inlet to bonnet. However, since it was desired to make comparisons with the actual test data shown in Fig. 4, actual temperature rises from test data were entered in Item 6.

7. *Density of air in the stack, pounds per cubic foot.* The densities of the air at each register were determined from the temperatures listed in Item 6 and an assumed barometric pressure of 29.40 in. of Hg.

8. *Air volume flowing corrected for density, cubic feet per minute.* Assuming that a constant mass flow rate is maintained, the volume flow rate corresponding to the density of Item 7 was calculated as follows:

$$\text{(Item 8)} = \frac{\text{(Item 5) (density of air entering furnace)}}{\text{(Item 7)}}$$

9. *Loss per 100 ft of branch duct, from friction chart, inches of water.* This is obtained from the friction chart[5] for the 6.6 in. equivalent diameter and the air quantities listed in Item 8. The friction chart is based on standard air of 0.075 lb per cu ft density.

10. *Loss per 100 ft of branch duct, corrected for density, inches of water.* As stated in THE GUIDE[6], "For ordinary ventilating work, friction may be assumed to vary directly as the density without serious error . . ." In the example

$$(\text{Item } 10) = (\text{Item } 9)\ \frac{(\text{Item } 7)}{(\text{Standard density})}$$

11. *Pressure to deliver corrected volume, inches of water.* This pressure is for the equivalent lengths of duct:

$$(\text{Item } 11) = \frac{(\text{Item } 10)\ (\text{Item } 2)}{100}$$

12. *Stack effect, inches of water.* The stack effect may be determined from the following equation:

$$(\text{Item } 12) = 7.50\ L \left(\frac{1}{T_s} - \frac{1}{535}\right)$$

where

L = height of stack above plenum, feet.

T_s = temperature of the air in the stack, Fahrenheit, absolute.

In the following section this stack effect has been considered to be the same as the effect produced by an exhauster fan mounted on the register of each branch.

13. *Combined stack pressure, inches of water.* The detailed procedure for determining the combined stack pressure is illustrated in Table B. The tabulated stack effect (b, c, d, and e) shown in Section A of Table B was obtained from Item 12 of Table A.

The first step shown in Section B of Table B was to treat the system as if the stack effect alone were acting. It was assumed that the stack effect was analogous to the effect of a fan mounted at the outlet of each of the stacks and exhausting from the plenum. The capacity of the fan would increase in proportion to the effective height of the stack and the temperature of the air.

Therefore, the stack effects of branch ducts Nos. 2 and 4 would be equivalent to fans of equal capacity since they were of the same height, the stack effect of branch duct No. 3 would be equivalent to a smaller fan and the stack effect of branch duct No. 1 would be zero and would have no fan since it was a basement outlet with no effective stack height.

Since the fans which were assumed to be acting at the outlets of branch ducts Nos. 2, 3, and 4, would be acting in parallel, the following excerpt from a section on Fans in Parallel[7], was used as a criterion.

"As in the case of fans in series the electric circuit analogy may be used also for fans in parallel. When two or more identical low pressure fans are placed in parallel the characteristic curve of the combined set is such that the capacity is the sum of the separate capacities for a given static or total pressure. . . . . .

"If the two fans are of different types or of different speeds or both. . . . . . Constant static pressure is used in this case upon which to combine capacity. The total pressure of the fan is based upon total capacity and total outlet area. In most cases this will be the average of the separate total pressures. If the fans lead into a common duct with different velocity pressures and in such a manner that they will mutually influence each other the static (pressure) at the outlet of one fan will drop and the other fan rise but with little change in total air quantity. Obviously, near the shut-off condition if one fan develops greater pressure than the other, the pressure curve in this region will be a compromise with air being forced back through the fan of lower pressure and the fan of higher pressure never attaining its maximum condition for this reason."

Using the foregoing analysis, the pressure in the plenum resulting from stack effect could be calculated if the branch duct temperatures were known. In the field these temperatures must be predicted if the method is to be of any value, but for the purpose of this analysis the known stack temperatures from test data were used.

TABLE B—DETAILED PROCEDURE FOR CALCULATING ITEM 13 OF TABLE A

BRANCH DUCT NO.	1	2	3	4
A. Stack Effects (Assuming no Blower Operation)				
Stack Pressure, combined	(l)	(m)	(n)	(o)
Stack Effect	0 (b)	0.014 (c)	0.008 (d)	0.016 (e)
Blower Pressure (a)				
Averaged Stack Effect	(h)	(g)	(f)	
Trunk Pressure	(i)	(j)	(k)	
B. Average Stack Effects in Trunk Duct (Assuming no Blower Operation)				
Stack Pressure, combined	(l)	(m)	(n)	(o)
Stack Effect	0 (b)	0.014 (c)	0.008 (d)	0.016 (e)
Blower Pressure (a)				
Averaged Stack Effect	0.013 (h)	0.012 (g)	0.016 (f)	
Trunk Pressure	(i)	(j)	(k)	
C. Combined Stack Pressure for Stack No. 1				
Stack Pressure, combined	0.024 (l)	(m)	(n)	(o)
Stack Effect	0 (b)	0.014 (c)	0.008 (d)	0.016 (e)
Blower Pressure 0.037 (a)				
Averaged Stack Effect	0.013 (h)	0.012 (g)	0.016 (f)	
Trunk Pressure	0.050 (i)	(j)	(k)	
D. Combined Stack Pressures for Stacks Nos. 2, 3, and 4				
Stack Pressure, combined	0.024 (l)	0.052(m)	0.040 (n)	0.056 (o)
Stack Effect	0 (b)	0.014 (c)	0.008 (d)	0.016 (e)
Blower Pressure 0.037 (a)				
Averaged Stack Effect	0.013 (h)	0.012 (g)	0.016 (f)	
Trunk Pressure	0.050 (i)	0.048 (j)	0.056 (k)	

Note: Blower Pressure (a) is from item 11 of Table A. Stack Effect (b), (c), (d), and (e) is from item 12 of Table A.

Based upon the preceding discussion and the assumption that stack effect acted as a fan, the following procedure was developed for predicting the unbalance of air distribution to the branch outlets under heated flow conditions. Starting with the last take-off, it was assumed that the suction in the trunk section between take-off 3 and take-off 4 was due to suction in branch duct No. 4 only, that is, suction (e) Table B is equal to suction (f). The suction between take-off 2 and take-off 3 was the average of the suction at take-off 3 and the suction in the trunk section between 3 and 4, in other words,

$$(g) = \frac{(d) + (f)}{2}$$

In like manner

$$(h) = \frac{(g) + (c)}{2}$$

Items (f), (g) and (h) have been designated as *averaged stack effect*.

In Section C of Table B the blower pressure (Item 11) from Table A, entered at (a) was superimposed upon these average stack effects. At take-off No. 1 in the trunk duct, the blower pressure of 0.037 in. was acting upstream from the take-off, and was tending to *push* the air out of branch duct No. 1. The averaged stack effect

(h) of 0.013 in. was opposed to the branch duct stack effect (b) of 0 in., and was tending to *pull* the air back into the trunk duct. The resulting total motive head in branch duct No. 1 has been designated as the combined stack pressure (1) and amounted to 0.037 − 0.013+0, or 0.024 in. By a similar analysis the downstream trunk pressure (i) is 0.037+0.013 − 0, or 0.050 in.

The values shown in Section D of Table B are the combined stack pressures for branch ducts 2, 3 and 4, and were obtained in the same manner. The values designated as (l), (m), (n), and (o) were then inserted in Item 13 in Table A.

14. *Stack pressure per 100 ft of branch, inches of water.*

$$(\text{Item 14}) = (\text{Item 13}) \times \frac{(100)}{(\text{Item 2})}$$

15. *Stack Pressure per 100 ft of branch, corrected for density, inches of water.* This correction is necessary in order to utilize the friction-chart as explained in (Item 10).

$$(\text{Item 15}) = (\text{Item 14}) \frac{(\text{standard density})}{(\text{Item 7})}$$

16. *Volume of heated air flowing, cubic feet per minute.* This was obtained from the friction-chart for the 6.6 in. diameter duct and the pressures of (Item 15).

17. *Volume corrected to inlet air temperature, cubic feet per minute.*

$$(\text{Item 17}) = (\text{Item 16}) \frac{(\text{Item 7})}{(\text{standard density})}$$

18. *Percent of total volume.* The total of the branch volumes listed in (Item 17) was 383 cfm.

19. *Branch air volumes for a total volume of 350 cfm.* Since the temperature rises entered in (Item 6) were based on a total air volume flowing of 350 cfm it was necessary to reduce proportionately the volumes given in (Item 17). That is, (Item 19) = (Item 17) $\frac{(\text{Item 18})}{(100)}$. The values listed in (Item 19) were plotted in Fig. 7 and indicated by the points shown for 25,000 Btu per hr input and 350 cfm.

The entire procedure given in this appendix was repeated for all the points shown in Fig. 7.

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1404

No. 1404

COMPARATIVE STUDY OF VENTILATING JETS FROM VARIOUS TYPES OF OUTLETS

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This paper is the result of research sponsored by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with *Case Institute of Technology.*

THEORY

THE GENERAL energy equation may be applied, in differential form, to any two sections of a discharge air stream. Difficulties are encountered in the integration across the stream, and in determining the constants to be used in a practical case. The discontinuities of a perforated panel, and the near impossibility of measuring the conversion of kinetic or mechanical energy into heat are further complications.

The degree of turbulence in the stream leaving an air outlet is affected by the pattern of perforations or the configuration of the grille-lattice. The manner in which the individual jets coalesce to form a homogeneous stream affects the loss of directional kinetic energy and the magnitude of the residual velocity in the direction of the axis of the stream. Since the total energy is so difficult to measure, the next best approach is to apply the laws of motion and of continuity, setting up an equation on the basis of conservation of momentum. The assumptions involved in setting up this equation are certainly violated by the actual stream conditions, especially those existing near the outlet and those near the end of the throw. In the intermediate zone, however, the dynamic or inertia forces prevail and there is ample experimental evidence that the momentum equation applies with reasonable accuracy[1].

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[1] Exponent numerals refer to References.

Presented at the Semi-Annual Meeting of THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Muskoka, Ontario, Canada, June 1950.

Using Fig. 1 as a representation of the air flow from a grille or other practical outlet, the initial total momentum in all the individual outlet jets, M_oV_o, may be equated to the momentum at any stream cross-section where the dynamic forces prevail:

$$M_o V_o = \frac{\rho}{g} Q_o V_o = \int_0^{Q_x} \frac{\rho}{g} VdQ = \int_0^{A_x} \frac{\rho}{g} V^2dA$$

where

- M_o = mass flow rate through outlet.
- ρ = density, pounds per cubic foot.
- g = acceleration due to gravity.
- dQ = VdA.
- V_o = velocity at vena contracta.
- Q_o = volume flow rate through outlet.
- V = residual velocity at any point in the stream profile at a normal distance X from the outlet.
- A_x = stream cross-sectional area at distance x.

For isothermal conditions, the density term ρ/g will drop out and the equation becomes:

$$Q_o V_o = \int_0^{A_x} V^2dA \quad \text{. . . . . (1)}$$

The original momentum Q_oV_o, can be evaluated from measurements made at or near the supply outlet, and the momentum at section X can be evaluated from the velocity profile at that section. (A velocity profile extending to zero in both directions is very difficult to obtain, see Figs. 2 and 3.) Wide use of Equation 1 is improbable because of the great amount of time required for accurate evaluation of velocity profiles in the various planes.

In a previous paper in this series[2] a substitute form of the momentum equation was derived using the more easily measured *maximum residual velocity* or centerline velocity of the jet. The same equation has been used by Rydberg and Norbäck[3] and others. Here again several assumptions are made, but experiment has shown that the equation gives a reasonably accurate measure of actual conditions in the stream from almost any type of opening, including perforated plates of low free area. In final form, the equation was expressed in terms of dimensionless parameters:

$$\frac{V_x}{V_o \sqrt{C_d R_{FA}}} = K \frac{\sqrt{A_c}}{X} \quad \text{. . . . . (2)}$$

This equation may also be written to give the throw in terms of cubic feet per minute and free area as follows:

$$X = \frac{KQ}{V_x \sqrt{A_f C_d}} = \frac{K V_f \sqrt{A_f}}{V_x \sqrt{C_d}} \quad \text{. . . . . (3)}$$

where

A_c = gross core area of outlet, grille face or perforated panel.
A_f = free open area of outlet = $A_c \times R_{FA}$.
C_d = coefficient of discharge for outlet.
R_{FA} = ratio of free area to gross core area (decimal).
K = the constant of proportionality or ***throw constant.***
Q = volume rate of flow through outlet.
V_f = Q/A_f = average velocity through free-open area.
V_o = velocity at vena contracta of jet from outlet or perforation (computed from $V = \sqrt{2gh}$).
V_x = maximum residual velocity at centerline distance X from outlet face.

Consistent units must be used in Equation 2; usually the most convenient are cfm, fpm and sq ft. For plotting on logarithmic coordinates with $V_x/V_o\sqrt{C_dR_{FA}}$

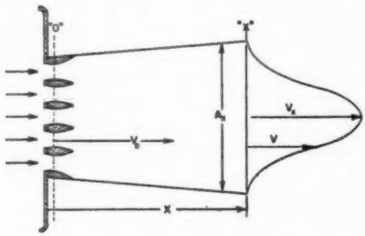


FIG. 1. AIR JET FROM A TYPICAL GRILLE OR PERFORATED PANEL

on ordinates, and $X/\sqrt{A_c}$ on abscissas (both dimensionless), the equation may be written:

$$\log \frac{V_x}{V_o\sqrt{C_dR_{FA}}} = -\log \frac{X}{\sqrt{A_c}} + \log K \quad . \; . \; . \; . \; . \; . \; . \; . \quad (4)$$

The log-log graph of Equation 4 should therefore be a straight line with a slope of -1 and intercept at $X/\sqrt{A_c} = 1$ numerically equal to K. Equation 3 is the same as the proposed Equation 1*a* in the 1944 paper of this series.[4]

The throw-constant K will be a maximum for a free-open rounded-entrance nozzle, indicating minimum losses of kinetic energy due to turbulence, and minimum angle of divergence of the stream. Either increased energy losses or increased angles of spread of the stream will give lower values of K. Experiments, however, show that this simple throw equation (Equation 3) is applicable and useful for predicting the behavior of free air streams from practically all outlets of the grille or panel type now in common use, including slots with an aspect ratio of less than 50.

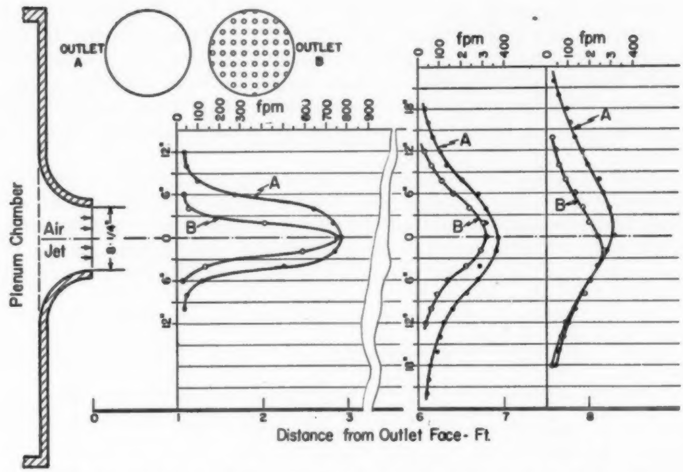
When it is required to determine the character of the stream very near the outlet, or at such a great distance from it that the viscous forces predominate, Equation 2 and the usual throw constants K of Table 2 do not apply.

TABLE 1—SUMMARY OF TEST DATA AND RESULTS

Item No.	Type of Outlet	Per Cent Free Area	See Fig. No.	No. of Test Pts.	Outlet Velocity Range fpm	Coefficient of Discharge C_d	Instrument Used for V_x[a]	Apparent Value of K Eqs. 2 and 3
	Tests at Case Institute, 1948–9							
1	Rounded nozzle, 8¼ in. diam	100	2,6	16	700–1700	0.98	A, B	7.0
2	Rounded nozzle, 2½ in. diam	100		5	900	0.93	A	7.0
3	Sharp-edged orifice, 4¼ in. diam	100		8	4400	0.73	A	7.0
4	Perforated metal panel, 8¼ in. diam	40.5	7,8	18	3800–4800	0.83	A	6.2
5	Perforated metal panel, 8¼ in. diam	9.13	2,6	7	2900	0.75	A, B	4.5
6	Perforated metal panel, 12 × 12 in. sq.	6.3		8	700–3300	0.74	B	3.7
7	Perforated metal panel, 24 in. diam	3.14		5	3700	0.79	A	2.7
	Tests at Case Institute of Technology, 1943							
8	Rounded nozzle, 11⅞ in. diam	100		29	1400–3900	0.99	C, E	7.0
9	Rounded nozzle, 8¼ in. diam	100		35	1300–7000	0.98	C	7.0
10	Rounded nozzle, 6 in. diam	100		41	1300–7000	0.98	C	6.5
11	Sharp-edged orifice, 11⅞ in. diam	100		28	1300–7000	0.60	C, E	6.7
12	Sharp-edged orifice, 8¾ in. diam	100		37	1600–7100	0.60	C	6.5
13	Sharp-edged orifice, 2.54 in. diam	100		16	3000–7000	0.60	C	5.8
14	Square opening 10 × 10 in., rounded	100		26	100–1600	0.83	C	5.3
15	Square opening 8 × 8 in., rounded	100		22	500–1600	0.88	C	5.4
16	Bar grille, 12 × 6 in., straight flow	84		40	400–2000	0.66	C	6.0
17	Bar grille, 20 × 4 in., straight flow	74		41	400–2500	0.78	C	5.0
18	Bar grille, 20 × 6 in., straight flow	72		27	400–1600	0.78	C	4.8
19	Bar grille, 12 × 6 in., 40° total spread	84[b]		23	900–2500	0.67	C	3.5
20	Bar grille, 20 × 6 in., 40° total spread	72[b]		20	1000–1500	0.75	C	4.0
21	Bar grille, 12 × 6 in., 40° total spread	67[b]		22	500–2400	0.63	C	3.2
22	Bar grille, 20 × 4 in., 40° total spread	74[b]	9	40	1200–2100	0.70	C	3.0
23	Bar grille, 12 × 6 in., 60° total spread	67[b]		31	700–2400	0.61	C	2.3
24	Bar grille, 20 × 6 in., 60° total spread	72[b]		11	1000–1900	0.74	C	2.2
25	Bar grille, 12 × 6 in., 60° total spread	84[b]		12	900–2500	0.62	C	2.6
26	Bar grille, 20 × 4 in., 60° total spread	74[b]	10	63	1200–2300	0.70	C	2.4
27	Bar grille, 20 × 6 in., 90° total spread	72[b]	11	33	1100–1900	0.72	C	1.8
	Tests at Kansas State College							
28	Nozzle, 2 in. diam	100	13	18	1600–2000	1.00	F	6.5
29	Nozzle, 4 in. diam	100	13	75	900–2000	1.00	F	6.5
30	Nozzle, 5.75 in. diam	100	13	12	1300	1.00	F	6.5
	Tests at Tech. University of Denmark							
31	Short nozzle, 3.9 in. diam	100		60	900–4900	0.87	F	7.1
32	Short nozzle, 2 in. diam	100		38	1800–5600	0.99	F	7.6
33	Conical nozzle, 5 in. diam	100		21	5400	0.98	F	7.2
34	Sharp-edged orifice, 1.9 in. diam	100	14	38	2500–5400	0.64	F	7.8
35	Sharp-edged rectangular orifice 2 × 4 in.	100	14	47	3200–5100	0.66	F	8.1
36	Open-end pipe, 11.8 in. diam	100		43	700–1200	0.94	F	7.2
37	Open-end duct, 11.8 × 11.8 in.	100		21	900	0.89	F	6.8
38	Double curved nozzle 7.86 in. diam	100	14	45	800–2300	0.97	F	7.0
	Tests at Chalmers Tech. University, Sweden							
39	Perforated panel, 8.8 × 8.8 in.	37.4	15	11		0.72	F	3.7
40	Perforated panel, 12.8 × 12.8 in.	37.4	15	11		0.72	F	3.7
41	Perforated panel, 19 × 19 in.	37.4	15	12		0.72	F	3.7
42	Perforated panel, 8.5 × 8.5 in.	19.6	15	32	1700–2300	0.67	F	4.2
43	Perforated panel, 8.5 × 8.5 in.	4.9	15	17	5800	0.67	F	4.5

[a]Key to instruments used for measuring free stream velocity V_x:
- A: Hot-wire anemometer
- B: Heated thermometer anemometer
- C: Bridled vane anemometer
- D: Heated thermocouple anemometer
- E: Rotating vane anemometer
- F: Pitot tube

[b]A_f and C_d for all diverging grilles based on free area of equivalent straight-flow grilles.



Actual perforations, outlet B, approximately 0.10 in. diameter, on ¼ in. centers.

FIG. 2. COMPARISON OF VELOCITY PROFILES FROM TWO DIFFERENT OUTLETS OF SAME GROSS AREA (HORIZONTAL SECTION)

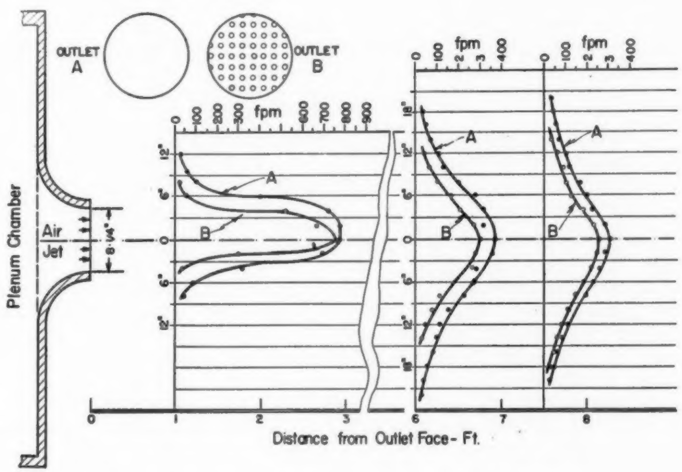
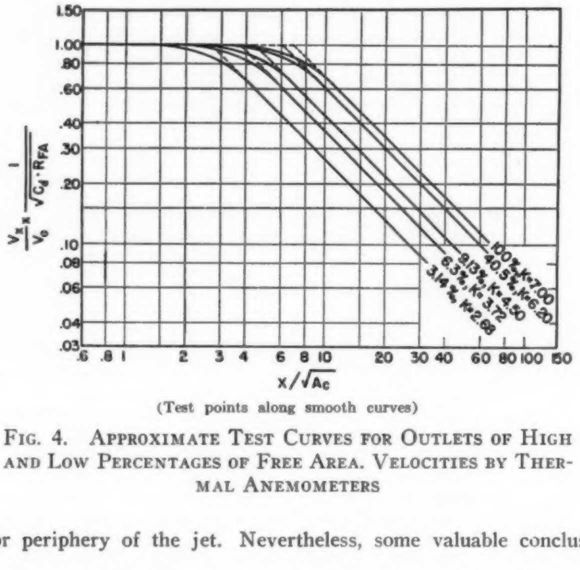


FIG. 3. SAME TEST AS FIG. 2 (VERTICAL SECTION)

TESTS OF NOZZLES AND PERFORATED PANELS

To test the assumption that Equations 1, 2 and 3 may be applied to straight-flow outlets of any type, irrespective of free area, a large number of experimental runs were made. In certain tests the overall size of the discharge outlet was held constant and the free area was varied. In other tests the gross area was varied. Runs were made over a wide range of air velocities and volumes. The results of some of the recent tests conducted are shown in Items 1 to 7 of Table 1.

To apply Equation 1, the stream velocity profiles must be determined at various distances from the discharge face. Figs. 2 and 3 show the velocity profiles obtained by horizontal and vertical traverses at distances of 1 ft, 6 ft, and 7.5 ft from the outlet. Actual experimental integration is most difficult because of the time required and the problem of extending the velocity curves to zero at the



(Test points along smooth curves)

FIG. 4. APPROXIMATE TEST CURVES FOR OUTLETS OF HIGH AND LOW PERCENTAGES OF FREE AREA. VELOCITIES BY THERMAL ANEMOMETERS

fringes or periphery of the jet. Nevertheless, some valuable conclusions are possible.

For the results shown in Figs. 2 and 3, the test conditions were adjusted to produce approximately the same maximum velocity at 1 ft from the discharge face, using two outlets of the same gross core area but of different free area. A long-radius flow nozzle of 8¼ in. diameter was used, first with free-open discharge, then covered by a thin perforated plate of only 9.13 percent free area. The initial momentum, Q_oV_o, for the perforated panel discharging 72.5 cfm was about 10 percent higher than the initial momentum, Q_oV_o, for the free-open nozzle discharging 263 cfm, when in both cases the velocity at the center of the jet, 1 ft from the face, was about 750 fpm. The velocity profiles at 1 ft, 6 ft, and 7.5 ft from the face all indicate, however, a higher momentum for the *free-open* outlet. The ratio of the areas under the two curves appears to remain

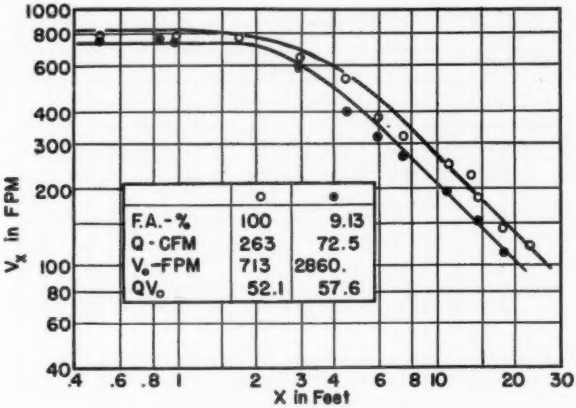


FIG. 5. MAXIMUM RESIDUAL VELOCITY IN STREAM, PLOTTING AGAINST DISTANCE

(Data from same tests as Fig. 2)

about the same, and this supports the assumption that the conservation of momentum applies in this zone of the free stream.

Applying Equation 2, if stream momentum could be equated between any two cross sections of the free and fully-formed jet (*i.e.* at 1 ft and 7.5 ft, Fig. 2), the value of K for all streams, regardless of the type of outlet, would be the same. But when the momentum at one of these sections is equated to the original momentum at the outlet (Q_oV_o), the turbulence imparted to the stream during the fusion of individual jets will result in a lower residual momentum at the distant cross-section. This will reduce the value of K in Equations 2 and 3 for a perforated panel. Fig. 4 represents one set of test data for outlets with

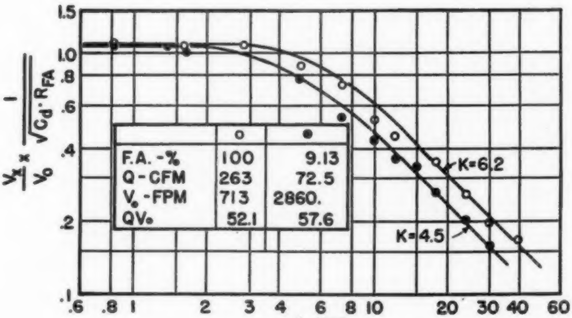


FIG. 6. DATA OF FIG. 5 CONVERTED TO DIMENSIONLESS COORDINATES

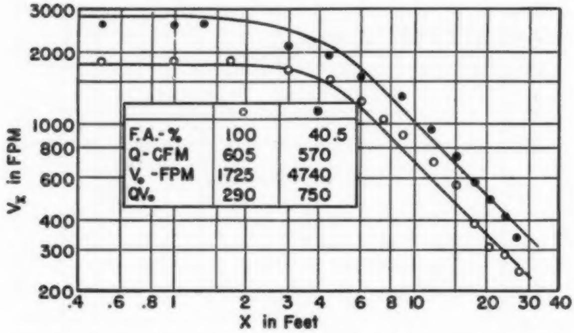


FIG. 7. MAXIMUM RESIDUAL STREAM VELOCITY FOR TWO OUTLETS OF THE SAME GROSS AREA DISCHARGING APPROXIMATELY EQUAL VOLUMES

different percentages of free area. It appears that the momentum loss due to turbulence is small if the free area of the outlet is 50 percent or more. The larger momentum loss for a panel of low free area is accompanied by a change in the length of the transition zone near the outlet, as shown.

Fig. 5 is plotted from the same tests as Figs. 2 and 3, but utilizing the velocity readings at the center of the stream only, *i.e.,* the maximum residual velocity at each cross-section. In Fig. 6 the data of Fig. 5 are plotted non-dimensionally, and the values of K are seen to be approximately 6.0 for the free-open nozzle and 4.5 for the perforated plate of 9.13 percent free area but the same gross area as the nozzle. (Read values at intersection $X/\sqrt{A_c} = 10$ and multiply results by 10.)

Both the velocity profiles of Figs. 2 and 3 and the maximum residual velocities

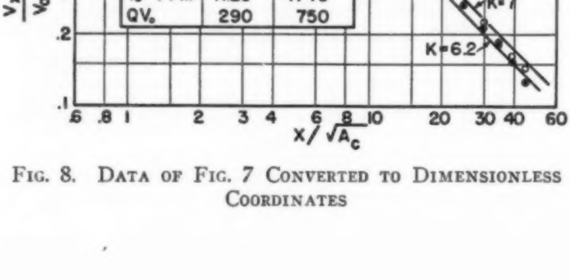


FIG. 8. DATA OF FIG. 7 CONVERTED TO DIMENSIONLESS COORDINATES

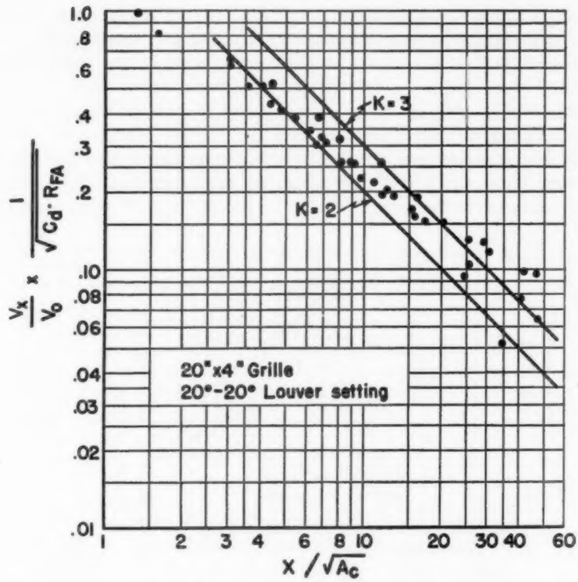


FIG. 9. TESTS ON GRILLE WITH DIVERGING VANES SET FOR 40 DEG TOTAL ANGLE OF SPREAD

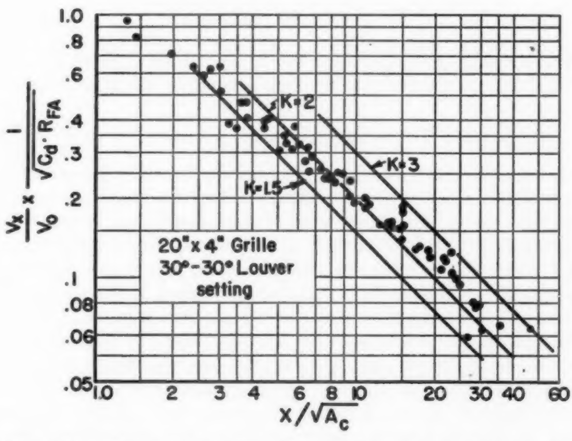


FIG. 10. TESTS ON GRILLE WITH DIVERGING VANES SET FOR 60 DEG TOTAL ANGLE OF SPREAD

plotted in Figs. 5 and 6 support the conclusion that a major loss of momentum occurs in the mixing of the individual jets from a perforated panel.

Fig. 7 shows that if air is to be discharged at a given volume-rate, it is entirely possible to use a perforated panel and secure a blow or throw that *exceeds* that from a free opening. The rate of discharge of primary air from the perforated panel of Fig. 7 was about 5 percent less than that from the free-open nozzle of the same (gross) area, but the momentum M_0V_0 of the air from the perforated plate was about 2½ times the momentum M_0V_0 of the air from free-open nozzle, hence the longer throw of the former. A non-dimensional plot

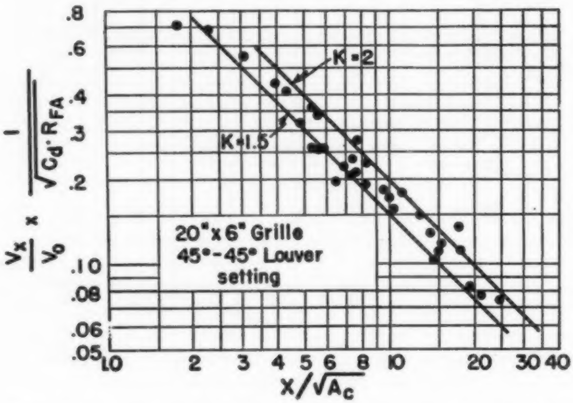


FIG. 11. TESTS ON GRILLE WITH DIVERGING VANES SET FOR 90 DEG TOTAL ANGLE OF SPREAD

of the same data, Fig. 8 shows values of K, Equation 2, about 7.0 for the free nozzle and 6.2 for the 40 percent free-open panel of the same gross core area.

TESTS OF DIVERGING GRILLES

When the louvers (vanes or bars) of an air discharge grille are set to produce a wide horizontal spread of the stream, the throw or length of the free stream is reduced. The reason for this is apparent from the derivation of Equation 2 (see Reference 2). Figs. 9, 10 and 11 give evidence that Equation 2 may again be applied, with reduced values of the throw constant K. Other tests are reported in Table 1. A comparison of stream envelopes is shown in plan view in Fig. 12. These three stream outlines were all obtained with the same grille (12 × 6 in.) with adjustable bars set first for straight flow, then for 40 deg total angle and finally for 60 deg total angle. In all calculations with diverging grilles, the free area A_f of the equivalent straight-flow grille has been used; hence V_f in Equation 3 is a fictitious quantity, used only for convenience.

The value of K to be used in any specific case is not easy to select, as shown from the several curves and from Table 1. There may be a long transition zone before the 45 deg portion of the curve is reached. The residual velocities are

in general *less* than would be indicated by solution from the momentum law in the form of Equation 2, using a constant value of K; hence such computed results will always be on the safe side. This is also true near the end of the throw (due to viscous forces).

CORRELATION AND COMPARISON OF TEST RESULTS

The results of many earlier tests have been published in the previous papers of this series.[1, 2, 4] Several other investigators have also published test results

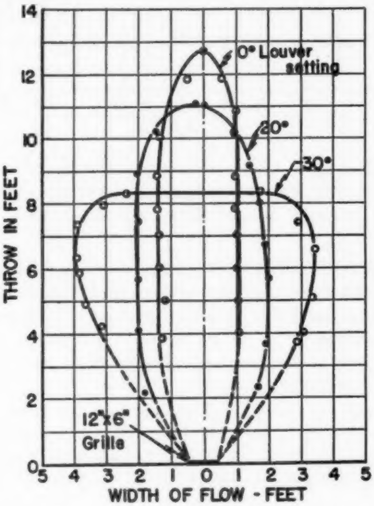


FIG. 12. TYPICAL STREAM ENVELOPES FOR THE SAME GRILLE WITH DIFFERENT SETTINGS OF ADJUSTABLE VANES. POINTS REPRESENT 100 FPM AXIAL VELOCITY

that confirm the assumption that Equation 2 is applicable to all outlets of the nozzle, grille or perforated panel type. A correlation of data from several sources is given in Table 1, and values of the throw constant K are indicated. Items 1 to 27 represent tests made at *Case Institute of Technology*. Items 28 to 30 refer to Fig. 13 giving unpublished data on three round nozzles tested at *Kansas State College* under the A.S.H.V.E. cooperative research project.[5] These latter tests, covering the downward projection of air at isothermal conditions, show a zone of nearly constant velocity to a distance of about $X/\sqrt{A_c} = 6$, then a sharp break and a 45 deg line with a value of K somewhere between 6 and 7. The range of initial velocities V_0 was more than 2 to 1, and most of the points for residual velocity V_x below 200 fall below the straight-line curves as previously explained (see open symbols at low end of curve).

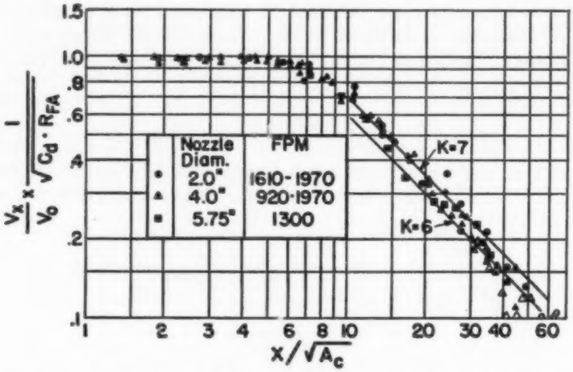


Fig. 13. Results Obtained at Kansas State College for Downward Projection of Unheated Air Streams from Three Rounded Nozzles

A large number of test results were obtained and published by Becher[6] covering a variety of round and rectangular orifices and nozzles at discharge velocities from 800 to 5500 fpm, see Table 1, Items 31 to 38. A number of Becher's test runs have been replotted in Fig. 14, using the dimensionless parameters of Equation 2. Here again is the constant velocity zone and the rather sharp break with a 45 deg line at approximately $K = 7$.

Similar results were obtained by Tenelius[7] on perforated panels of 5 to 37 percent free area, as shown in Fig. 15. Here the values of the throw constant K, are in the low range, roughly from 3 to 5, but the effect of free area on K, and the 45 deg slope, are not fully verified. Data in Fig. 15 shown as solid black

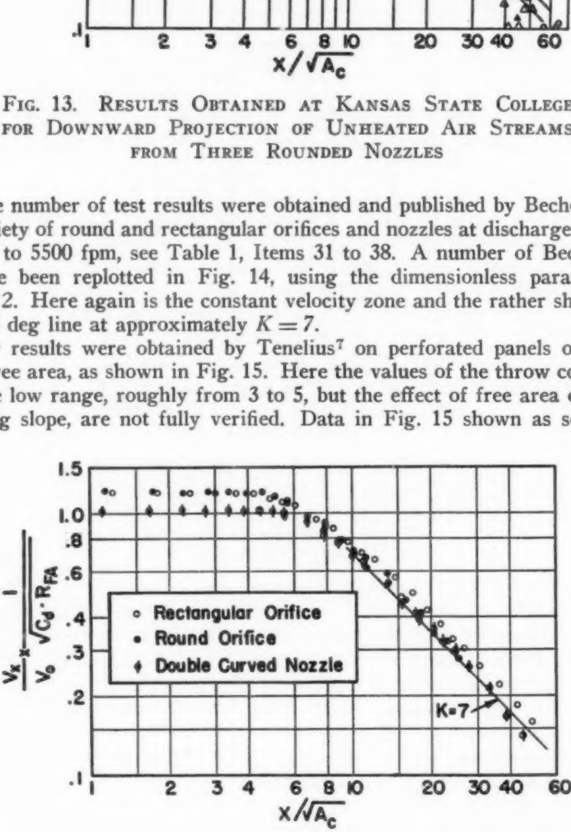


Fig. 14. Test Results Obtained at the Technical University of Denmark

points are for tests on three different sizes of panel all of 37.4 percent free area and identical hole spacing.

Results obtained by Rydberg and Norbäck[8] would be similar, and they suggested a value of the throw constant K of about 5.6 for *grids of various constructions.*

PRACTICAL APPLICATIONS AND CONCLUSIONS

The performance of the free air stream is obtainable by Equations 2 and 3, for round or rectangular orifices, nozzles, grilles, registers and perforated panels, also for slots having an aspect ratio less than 50, provided good entrance or approach conditions are maintained. The velocity *vs.* throw relationship involves two simple dimensionless ratios, a velocity ratio $V_x/V_o\sqrt{C_dR_{FA}}$ and a ratio

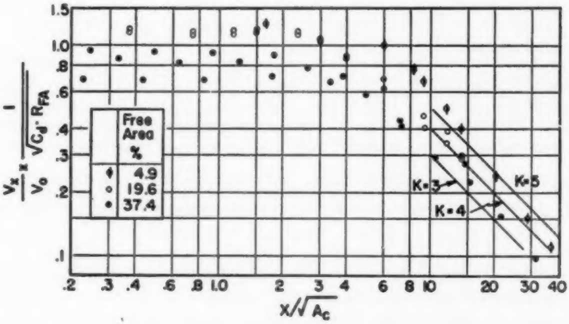


FIG. 15. TEST RESULTS ON PERFORATED METAL PANELS OBTAINED AT CHALMERS TECHNICAL UNIVERSITY, GOTHENBERG, SWEDEN

of linear dimensions $X/\sqrt{A_c}$. The selection of the constant of proportionality between these two ratios, *i.e.,* of the throw constant K, requires some judgment, and depends on the methods of measuring V_o and V_x. Thermal anemometers of the hot-wire, heated thermocouple or heated thermometer type will give slightly higher readings of V_x than mechanical directional anemometers (see discussion of experimental methods in Appendix.)

A suggested table of throw constants K for practical use, is given as Table 2. It is fully recognized that the use of Equations 2 and 3, with values of K from Table 2 has decided limitations including those imposed by differences in temperature. In the range from 8 to 50 diameters from the outlet (*i.e.,* $X/\sqrt{A_c}$ from 8 to 50) a reasonably good accuracy is usually obtained until the final velocity at the center of the jet has been reduced below 100 fpm. (The *average* velocity across the jet will then be well below 50 fpm.) For straight-flow outlets and perforated panels, the *constant velocity zone* usually extends at least 3 diameters from the face, but for diverging grilles it is often only one diameter in length. No simple method seems to be available for predicting the maximum stream velocity in the zone nearer the outlet, say from 2 to 8 diameters from

TABLE 2—SUGGESTED VALUES FOR THE THROW CONSTANT K. EQUATIONS 2 AND 3

TYPE OF OUTLET	VALUE OF K WHEN V_x IS MEASURED WITH THERMAL ANEMOMETER (V_x above 200)
Rounded-entrance nozzles (Round, square or rectangular)	7.0
Square-edged openings (Aspect ratio 1 to 25)	6.5
Long slots (Aspect ratio 35 to 50)	6.0
Grilles and registers (Free opening 60 to 90%)	5.5
Perforated panels (Free opening 25 to 40%)	5.0
Perforated panels (Free opening 3 to 12%)	4.0
Diverging grilles Total angle 40 deg	3.5
Diverging grilles Total angle 60 deg	2.5
Diverging grilles Total angle 90 deg	2.0

Notes:
1. For computations in the range $V_x = 75$ to $V_x = 175$, the values given in table may be 10 to 20 percent high.
2. When V_x is measured with a mechanical vane anemometer, the values given in table may be 5 to 10 percent high.

the face, except to assume that $V_x/V_o\sqrt{C_d R_{FA}} = 1$ (see Figs. 4, 6, 8, etc.). If Equations 2 and 3 and Table 2 are applied for this very near zone, or for the very far zone (V_x below 100), the computed results will always be high, *i.e.,* on the safe side.

Many refinements in the computation of free-stream performance are still to be made, but Equation 2 and the estimated values of Table 2 are suggested as good approximations, pending additional work on the many questions that are as yet unanswered.

ACKNOWLEDGMENTS

The authors of this paper are greatly indebted to C. Y. Young and Miss Ruth Ehrenfeld, graduate research assistants at *Case Institute of Technology,* for their months of careful work in observation, calculation and plotting of test data presented in this paper.

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APPENDIX

EXPERIMENTAL VERIFICATION OF EQUATIONS 1, 2 AND 3

The experimental investigations conducted in the summer of 1949 at *Case Institute of Technology* were concerned with the verification of Equations 1, 2 and 3 under isothermal conditions. To check Equation 1, horizontal and vertical velocity profiles were taken, as shown by the example of Figs. 2 and 3. Equation 2 was tested by determining centerline velocities at various distances from the outlet discharge face, as in Figs. 5 and 6.

Most of the velocity determinations were made with a commercial type electric hot-wire anemometer. For purposes of comparison, a heated thermometer was also used in evaluating centerline velocities for the data in Items 1 and 5, Table 1. Inasmuch as it took approximately 15 min to obtain a reading with the heated thermometer, and because it agreed with the indication of the hot-wire anemometer within 2 to 3 percent, use of the heated thermometer was discontinued for the remainder of the runs. (Table 1, Items 2, 3, 4, and 7.) The calibration curves of the heated thermometer are shown in Fig. 16; they agree very closely with those in a previous investigation[2].

The hot-wire instrument consisted of a Wheatstone bridge with one of its resistances heated electrically. Variations of air velocity across this heated resistance change its value. Damping characteristics were quite pronounced and facilitated the taking of readings in the air stream. The calibration curves are given in Fig. 17.

The calibration procedure for the velocity measuring instruments was similar to that already described in previous papers in this series. Checks were obtained by using both flow nozzles and thin-plate orifices as the primary metering devices. Calibration results from the use of orifices and flow nozzles of various sizes usually checked within 1 percent of each other.

The various outlets and perforated panels were mounted interchangeably on a 48 in. cubical plenum chamber. The proportions and arrangement of the supply fan, duct and plenum corresponded to those specified in the Standard Test Code for Centrifugal and Axial Flow Fans*. Inclined manometers, calibrated against a hook gage, were used to obtain the pressure drop across a standard orifice in the duct, and to measure the static pressure in the plenum.

For better duplication of approach conditions the 8¼ in. nozzle was used for several runs. Variations in free area were attained by mounting the perforated panels on the discharge face of the nozzle. The other runs were made with the panels mounted directly in the wall of the plenum.

The velocity measuring instrument was mounted at the end of a long steel rod, and the instrument could be centered in any one of the squares of a 7-ft vertical grid.

* Bulletin No. 103 *National Association of Fan Manufacturers*.

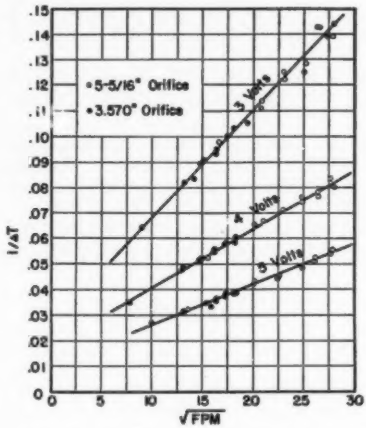

FIG. 16. CALIBRATION OF HEATED THERMOMETER NO. 56 AT VARIOUS IMPRESSED VOLTAGES

(ΔT temperature difference between heated and unheated thermometer)

The grid was formed by 2 in. squares of fine thread; it was centered on the axis of the outlet stream at various distances from the discharge face.

The location of the experimental setup in the laboratory was such that it was free from drafts and obstructions. The distance from the outlet to the opposite wall was

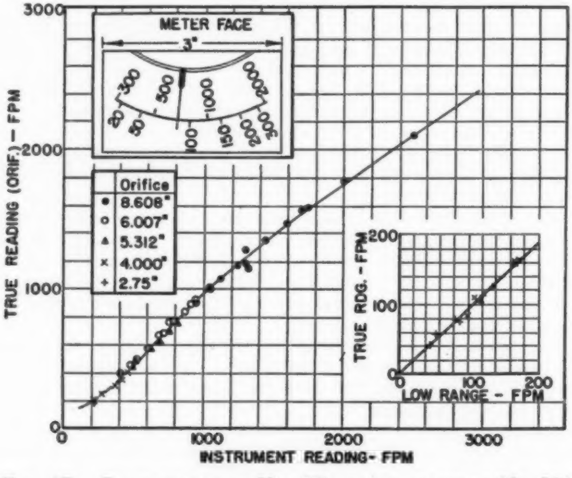

FIG. 17. CALIBRATION OF HOT-WIRE ANEMOMETER NO. 781

30 ft; the nearest wall on the side was 15 ft from the stream centerline; the overhead was about 15 ft above the centerline, and the center of the outlet was 5 ft from the floor.

DISCUSSION

LINN HELANDER, Manhattan, Kans. (WRITTEN): The authors have presented an excellent, simple correlation of data on center line velocities of isothermal jets projected from orifices of various types, including perforated plates. Their conclusions have practical value and theoretical significance. A formula of the type given by the authors in Equation 2 can be developed analytically for that portion of a jet where the velocity profiles are invariable in form from cross section to cross section. These profiles have a pattern such that they may be consolidated into a single curve when replotted in terms of appropriate dimensionless parameters. One of these parameters is V_a/V_x, the ratio of the axial velocity of a point in the jet to the velocity at the axis in the cross sectional plane of the point. The other is r/x, the ratio of the radial and axial coordinates of the point, referred to an origin on the axis, not necessarily in the plane of the orifice. The analytically developed equation has the form:

$$\frac{V_x}{V_0} = \frac{K\sqrt{A_0}}{X + X_0}$$

where

$$X_0 = \left(\frac{b\,D_0}{ca}\right) - X_i$$

b and c are defined by the relationship:

$b\,M_0V_0 = cMV_x$ = momentum rate of flow in zone of uniform velocity pattern.

X_i = axial distance from orifice to the plane that marks the beginning of the zone of uniform velocity pattern.

D_0 = orifice diameter $= 2\sqrt{A_0}/\sqrt{\pi}$

$$K = 2\left(\frac{b}{c}\right) \Big/ \left[a\sqrt{\pi}\right]$$

where

a = rate of room air entrainment in zone of uniform pattern of flow, expressed in pounds of room air per pound of orifice air per orifice diameter of jet travel.

From Fig. 4 in the paper,

$K = 7.00$ when b is unity.

Using this value of K in the preceding equation for K and solving for ac, we find

$ac = 0.16$ approximately.

This value is within the range of values for rounded nozzles obtained by experiment.

The authors have shown that for the types of orifice covered in their paper, X_0 may be equated to zero. This is a fortunate coincidence, particularly so in view of the large variety of orifices investigated. However, it is a result that may not have universal validity. It may not apply, for example, to jets emitted from outlets that produce, at the orifice, a velocity profile which is approximately parabolic. When X_0 is zero:

$$X_i = \left(\frac{bD_0}{ca}\right)$$

Setting b equal to unity and employing the value of ac evaluated above, we obtain

$X_i = 6D_o$, approximately, for rounded nozzles.

This value of X_i is in conformity with values that have been indicated by other investigators.

Equation 2 in the paper implies that if a, the rate of room air entrainment per orifice diameter of travel per pound of orifice air, is a constant in the principal zone, it cannot be a constant of the same magnitude in the primary zone of the jet; that is, in the zone between the orifice and the beginning of the zone of uniform pattern of velocity profile. Designating by a_o, the average value of a in the primary zone, and making use of the relationship, $b\ M_oV_o = c\ M_iV_{x_i}$ with V_{x_i} replaced by V_o, we obtain,

$$a_o = \left[1 - \left(\frac{c}{b}\right)\right] a$$

Assigning to c a value between 0.5 and 0.6 and to b a value of unity, we get for a_o a value that lies between $0.4a$ and $0.5a$. Only meager experimental data are available on the manner in which the rate of mass flow varies in the primary zone with distance from the orifice. However, such data as are available support the reasonableness of the above evaluation.

The authors point to obstacles in the way of applying the energy equation to the flow of jets and instead of using that equation, they rightly employ the momentum equation. However, for that section of the jet wherein the velocity pattern is uniform in shape, one can readily show that

$$\frac{M\overline{V}^2}{\overline{M}_i\overline{V}_i^2} = \frac{X_i + X_o}{X + X_o} = \frac{M_i}{M}.$$

This equation leads directly to an equation of the form,

$$\frac{\overline{V}}{\overline{V}_i} = \frac{V_x}{V_o} = \frac{X_i + X_o}{X + X_o}$$

The authors report a substantial loss of momentum in the vicinity of the orifice when a perforated plate is employed as the orifice. Velocities at the orifice were determined from measured differences between the pressure in the plenum chamber and that of the atmosphere. In view of this, the loss of momentum reported would seemingly be due to either boundary forces or to the resistance offered by the perforated plate itself. Boundary forces could exist, particularly if the ambient air in the vicinity of the orifice has a velocity opposed to that of the jet. However, a more likely explanation of the loss of momentum would seem to be the resistance to flow offered by the perforated plate.

From the foregoing, it will be obvious that a good order of consistency exists between the formula for center line velocities developed by the authors and characteristic data on other aspects of the pattern of flow of isothermal jets than those directly covered in the paper. It will also be obvious that important conclusions regarding the behavior of jets can be derived from data on center line velocities. Of particular importance in the analysis of such data is the point X_i, located in the principal plane of the jet, this plane marking the beginning of the zone in which the axial velocity profiles are homologous in form. Regardless of the type of orifice,

$$\frac{V_x}{V_o} = \frac{X_i + X_o}{X + X_o}$$

provided only that the principal zone of the jet has velocity profiles that are symmetrical and identical in shape when plotted in terms of appropriate dimensionless parameters.

J. N. LIVERMORE, Detroit, Mich. (WRITTEN): I wish to emphasize that fundamental work on *free* isothermal air streams such as is presented in this paper, is an essential step in arriving at reliable working data on the behavior of *non*-isothermal air streams. Of particular concern are those air streams below room temperature, which are not free streams but are influenced by walls, ceilings, floors and obstructions. It seems to me that this paper, in pointing out its practical values, fails to state the most practical aspect of all: that the theory and working data it develops will ultimately become a part of a most badly needed design tool. I think we all await the day when air distribution in confined spaces ceases to be an art and becomes a science. The authors of this paper are to be commended for their contribution in this direction. Continued research on this subject until it applies directly to field conditions deserves much encouragement and support.

Relative to the data in this paper, I have the following questions on the K factors to be applied to adjustable bar grilles: (1) Have K factors been established for bar angles other than symmetrical settings, for example, for *all* bars turned toward one side of the grille? (2) Have K factors been established for bar angles in varying degrees of divergence from the center of the grille, for example, little or no angle at the outside? (3) Do the K factors presented apply to a double deflection grille in which the bars may spread the air stream both horizontally and vertically?

G. B. PRIESTER, Baltimore, Md. (WRITTEN): This paper is an excellent contribution, and comments on it from representatives of outlet manufacturers should be of interest to all.

What the application engineer would like to have is a simple chart or table to aid him in quickly selecting suitable outlets for common commercial air conditioning applications. The calculation procedure, of course, could be used for special cases.

LESTER T. AVERY, Cleveland, Ohio: I have studied this paper to determine if the material applies only to horizontal air flow. Apparently this is the case and presumably the temperature of the air is the same as that of the surrounding air. It may be of interest to others who have done practical work with the perforated panel to say that we have installed a great many systems using perforated panels, perforated ducts, perforated grilles and have tried to learn how to handle them properly. There are some very interesting phenomena which occur and we should get engineering data which are needed for practical work. For example, an orifice of the size they show will act as predicted if it is alone. But if you extend it into a continuous perforated grille it will act differently. We are at a complete loss to determine the throw when the perforations are close to the ceiling. Apparently in order for the throw to occur as predicted there must be a source of aspirated air all around. We found that if you install the perforated duct or grille close to the ceiling, the air tends to rise to the ceiling and the velocity is lost immediately and does not cover the required area. By moving the duct lower and away from the back wall, we permit aspirated air to come in on the top as well as the bottom and get the expected throw and distribution. It seems very important in planning aspiration with any kind of an outlet to provide ample space for aspirated air to get to the primary jet.

Another problem which I hope will be investigated is what happens to the ceiling distribution where you use the complete perforated ceiling and the air comes down uniformly all over the space. Is there aspiration taking place and if so, how? What velocities of air through the orifices are best to get uniform aspiration and mixing?

I think this paper helps in its way to understand the problem, but there is much to be done in order to complete our knowledge on the subject of jets and orifices.

AUTHORS' CLOSURE (G. L. TUVE): We are glad that President Avery and Mr. Livermore have indicated the complexity of practical problems in air-supply jets, and hence the limitations of present knowledge. The data in this paper were obtained with

isothermal streams, and covered only the behavior of jets from single grilles or panels. The largest panel we used was four square feet in area, and we agree with Mr. Avery that data on much larger panels are needed. The adjustable-bar grilles used in our tests were set to give uniform fan-shaped streams of 40-60- and 90-deg horizontal angles, with no vertical deflection. Our main point is that the velocity along the centerline of the jet can be computed in similar manner, whether the outlet be an orifice, nozzle, grille, slot or perforated plate, over the range of 3 percent to 100 percent free area.

The results presented are not necessarily confined to horizontal air flow. For instance, the data obtained by Professor Helander, and shown in Table 1 and in Fig. 13 are for downward projection of air. But these data as well as those obtained in Denmark and in Sweden, all agree with our findings as regards the projection or throw of the air stream.

The Technical Advisory Committee on Air Distribution has requested data on ceiling outlets, hence we hope that in the near future we will be able to answer Mr. Avery's questions about a stream projected along the ceiling. We agree that more data are needed on very long slots.

Professor Helander's comments indicate a general agreement with our findings for the *zone of uniform velocity pattern*. We must admit that we do not fully understand why the theory of conservation of momentum predicts the equations so closely when careful measurements show successive reductions in the total momentum. Figs. 2 and 3 emphasize the greater loss of momentum near the outlet face in the case of a perforated panel. I do not see why the resistance to flow offered by the perforated plate should affect the momentum, but we do know that the pressure varies slightly from point to point across the panel and close to the discharge face.

1405

No. 1405

ENERGY LOSSES IN 90-DEGREE DUCT ELBOWS†

A Survey and Analysis of Available Information

By D. W. Locklin*, Cleveland, Ohio

This paper is the result of research carried on by THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at its Research Laboratory located at 7218 Euclid Ave., Cleveland 3, Ohio.

As part of the program of the Technical Advisory Committee on Air Distribution††, and the Subcommittee on Air Duct Friction, a survey is being made of available information on energy losses in duct fittings. The scope of the survey is confined to losses in through-flow duct fittings which provide for change in direction of the air stream; included are losses in vaned elbows, compound elbows, and certain special elbows. It is the purpose of this survey to assemble the available data, estimate the reliability and application of these data, suggest data which may be acceptable for use in design, and determine areas in which additional investigation seems desirable.

This paper covers flow energy losses in single duct elbows of 90 deg, assembled in duct systems and having substantially uniform cross-section throughout the bend. Attention has been given to the presentation of results in a manner which will facilitate their use in practical engineering calculations, with particular application to air duct design.

Nature of Flow in Duct Elbows

Flow Phenomena: The geometry of an elbow plays an important part in determining the nature of flow in it; however, certain characteristics of flow common to all elbows may be generalized.

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† Portions of this paper form part of a thesis submitted by the author to Case Institute of Technology in fulfillment of the requirements for the degree of Master of Science in Mechanical Engineering.

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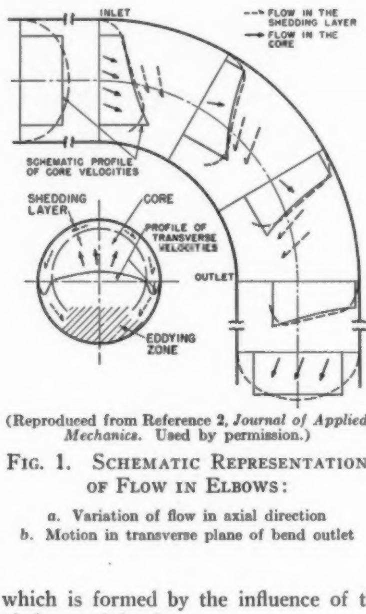
Presented at the Semi-Annual Meeting of The American Society of Heating and Ventilating Engineers, Muskoka, Ontario, Canada, June 1950.

As a result of his experimental investigation[1, 2] of velocity distributions in duct elbows, Weske describes three distinct regions of flow: (1) the core, (2) the shedding layer, and (3) the region of eddying flow. These regions are schematically shown in Fig. 1.

The flow in the *core* or central body of fluid is predominantly axial, transverse velocity components being small compared to components parallel to the axis of the duct. The velocity distribution frequently approximates a condition in which the velocity is inversely proportional to the radius of stream line curvature. This central body of fluid may be visualized as the *core* emerging from



(Reproduced from Reference 2, *Journal of Applied Mechanics*. Used by permission.)

FIG. 1. SCHEMATIC REPRESENTATION OF FLOW IN ELBOWS:

a. Variation of flow in axial direction
b. Motion in transverse plane of bend outlet

the boundary layer which is formed by the influence of the elbow in altering the velocity profile of the straight duct.

The *shedding layer* includes the fluid near the duct wall in which the velocity components axially and normal to the wall are small, and the component tangent to the core is large. The peripheral flow in this region is directed toward the inside of the bend as shown in Fig. 1, serving to shed the fluid of lower total energy near the wall.

The two currents of peripheral flow from the shedding layer impinge at the inside of the bend in the *region of eddying flow.* This region is occupied by turbulent fluid of lowest total energy resulting from the sorting of fluid particles according to energy content. Backflow may occur in the eddying region for elbows of small radius ratio. The flow in the shedding layer and its circulation through the core is commonly referred to as *secondary* or *vortex* flow.

[1] Exponent numerals refer to Bibliography.

NOMENCLATURE

Dimensional Terms

A_1 and A_2 = area of bend at entrance and exit, respectively; square feet.
c = velocity of sound in the fluid, feet per second.
D = duct hydraulic diameter, ft (or inches where noted); $D = 4 \times \frac{\text{area}}{\text{perimeter}}$.

For rectangular ducts: $D = \frac{2W}{1 + W/H}$

For round and square ducts: $D = W$.

g = acceleration of gravity, feet per (second) (second).
h = total head loss due to a fitting, feet of fluid flowing; $h = \Delta p/\rho$.
H = duct dimension perpendicular to plane of bend, feet (see Fig. 2).
l = length of duct, feet.
l' = curved centerline length of elbow, feet (for 90 deg elbows: $l' = \pi R/2$).
L = length of duct, in addition to length of m-o-n of Fig. 2, imposing resistance equivalent to that of a given elbow, feet; $L = L_t - 2R$.
L_t = total length of duct imposing resistance equivalent to that of a given elbow, feet.
Δp = total pressure loss due to a fitting, pounds per square foot.
$\Delta p'$ = net pressure loss due to a fitting, pounds per square foot.
R = centerline radius of bend, feet (see Fig. 2).
V = velocity of fluid, feet per second.
W = duct dimension in plane of bend, feet (see Fig. 2).
ϵ = absolute roughness of duct system, feet.
μ = viscosity of fluid, pounds per (foot) (second).
ρ = density of fluid, pounds per cubic foot.

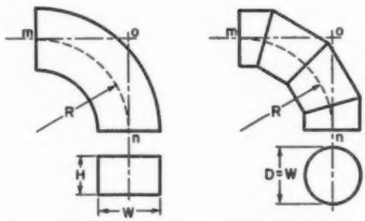
Dimensionless Terms

a = aspect ratio factor; the ratio of k/f at a given aspect ratio to k/f at unity aspect ratio; $a = \frac{k/f}{(k/f)_{(H/W=1)}}$.
f = friction factor in equation: $h_f = f \frac{l}{D} \frac{V^2}{2g}$.
H/W = aspect ratio.
k = total loss coefficient of elbow in equation: $h = k \frac{V^2}{2g}$.
k' = net loss coefficient of elbow; $k' = k - f\, l'/D$.
L/D = additional equivalent length in terms of hydraulic diameter.
L/W = additional equivalent length in terms of width.
L_t/D = total equivalent length in terms of hydraulic diameter; $L_t/D = k/f$.
L_t/W = total equivalent length in terms of width.
M = Mach number; $M = V/c$.
N_{Re} = Reynolds number; $N_{Re} = \frac{\rho D V}{\mu}$.
R/W = radius ratio.
ϵ/D = relative roughness.
θ = angle of bend in radians; $\theta = l'/R$.

The presence of eddying flow and these vortex pairs have been established by dye traces,[3] casts,[4] velocity surveys, [1, 5, 6, 7, 8, 9, 10, 11] and photographs. [6, 11, 12, 13]

Energy Losses: Losses of available energy occur due to shear stresses at the duct wall and within the fluid itself, not only in the flow at the bend but downstream from the bend in the reforming of the velocity distribution.

A number of attempts[3, 14, 15] have been made to predict energy losses analytically, though only the most important will be mentioned here. Theoretical investigations have been made by Dean[16] and Adler[5] for laminar flow in elbows, however these results cannot be extended to turbulent flow. Results of a qualitative nature were obtained by Nippert[6] who examined the losses due to secondary vortex flow by analogy to shearing stresses in a twisted bar. More recently



(a) Rectangular Elbow (b) 4 Piece Round Elbow

FIG. 2. TERMS DEFINING GEOMETRY OF ELBOW

Weske[2] has presented evidence to indicate that the loss is proportional to the quantity of flow in the shedding layer. Further, he has treated the equation of motion in the shedding layer of elbows in round ducts, and has derived an approximation for the energy loss in terms of radius ratio, the upstream velocity distribution, and the pitch of the secondary flow. Since data are lacking for this pitch, the relation is indicative of trends only.

In the absence of a theoretical treatment which satisfactorily copes with the complexity of flow and the many elbow geometries encountered, it is necessary to resort to the empirical correlation of experimental results.

Several different methods have been employed in presenting experimental results, so that one of the problems of this survey has been to reduce these data to common terms. Some investigators have deducted from the total pressure loss Δp for the curved portion of duct, the loss for a length of straight pipe equal to the length of the bend axis and have termed the remainder $\Delta p'$ the ***deflection*** or ***shock*** loss. Use of the term *shock* in this sense and in the description of losses at a sudden enlargement is believed misleading.* The practice of deducting the friction loss for the axial length is not followed in this paper, since the complex nature of the flow makes the friction loss assigned to the

* In modern fluid mechanics, the term *shock* denotes the compression of fluid from a super-sonic velocity to a sub-sonic velocity.

axial length entirely fictitious. Furthermore, use of the total loss Δp is more conventional to heating and ventilating practice.

In the empirical approach, it must be borne in mind that the data represent many conditions of flow and several different test methods; no systematic and complete set of data is available.

EMPIRICAL APPROACH

Variables Affecting Loss: Fourteen dimensional variables are known to be significant in fluid flow through conduit bends. Expressed mathematically by the symbols of the nomenclature (see p. 481) and Fig. 2,

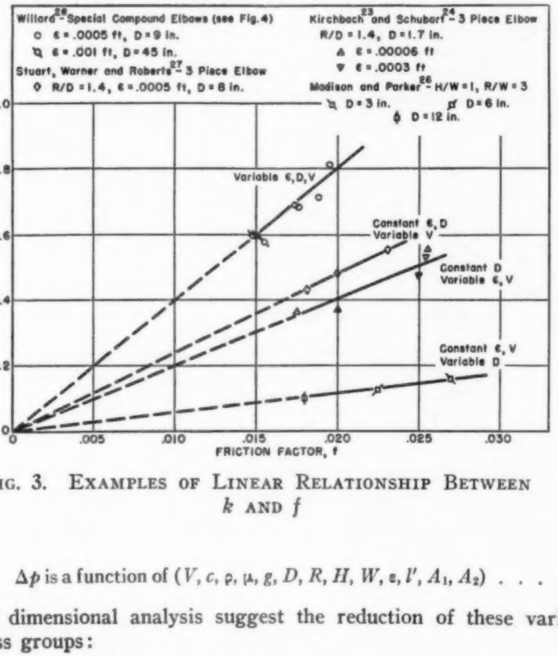


FIG. 3. EXAMPLES OF LINEAR RELATIONSHIP BETWEEN k AND f

$$\Delta p \text{ is a function of } (V, c, \rho, \mu, g, D, R, H, W, \varepsilon, l', A_1, A_2) \quad \ldots\ldots \quad (1)$$

Methods of dimensional analysis suggest the reduction of these variables to dimensionless groups:

$$\frac{\Delta p}{\rho\, V^2/2g} \text{ is a function of } \left[\frac{\rho\, V D}{\mu},\ V/c,\ R/W,\ H/W,\ \varepsilon/D,\ l'/R,\ A_1/A_2\right] \quad \ldots \quad (2)$$

or

$$k \text{ is a function of } (N_{Re},\ M,\ R/W,\ H/W,\ \varepsilon/D,\ \theta,\ A_1/A_2)$$

where

k = total loss coefficient of the elbow.

Inasmuch as this paper deals with 90-degree elbows of constant area, the terms for area ratio A_1/A_2 and angle θ are dropped. For the velocities encountered

in air conditioning practice, compressibility effects are negligible within a fitting so that the Mach number M need not be considered.

The Friction Factor: In an attempt to further reduce the number of variables and thus to simplify the empirical approach, the possibility was investigated of replacing the relative roughness ϵ/D and Reynolds number N_{Re} by the friction factor f for straight pipe. The relation of these factors has been offered by Moody[17] in the form of the equation

$$f = 0.0055\,[1 + (20{,}000\,\epsilon/D + 10^6/N_{Re})^{1/3}] \quad . \; . \; . \; . \; . \; . \; . \; . \; . \quad (3)$$

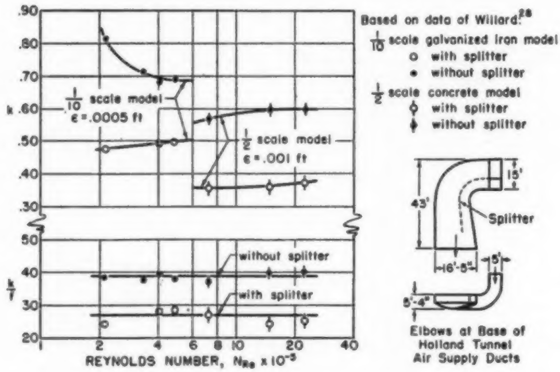


FIG. 4. LOSS DATA FOR MODELS OF HOLLAND TUNNEL ELBOWS, ILLUSTRATING USE OF k/f WHEN SIZE ROUGHNESS, AND VELOCITY ARE VARIED

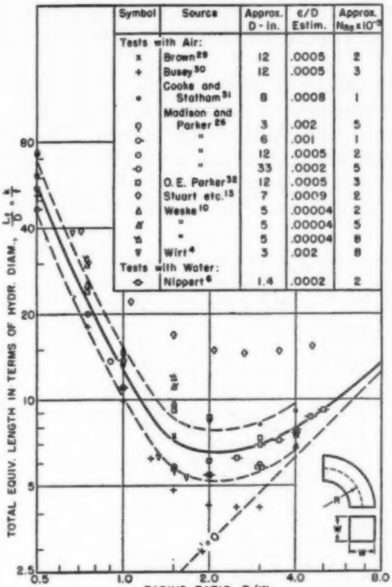
Rouse[18] has shown that for fully developed turbulence in straight pipes, [illegible] velocity distribution is dependent only upon the friction factor; it is for the purpose of characterizing the flow at a given combination of N_{Re} and ϵ/D that the friction factor is introduced. Thus

$$k \text{ is a function of } (f,\ R/W,\ H/W) \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (4)$$

It has been established by Weske[10] and by Huebscher[19] that for reasonably smooth ducts the friction factor is independent of aspect ratio up to 8:1. The roughness of an elbow in general corresponds to that for straight ducts, the loss due to elbow joints being included in the loss coefficient k.

It was observed by Abramovich[20], in analyzing the results of several European investigations,[21-25] that the net bend loss coefficient k' varied linearly with the friction factor f for elbows in which backflow was not appreciable. The variation in friction factor for these tests was afforded by varying the velocity and by roughening the inside of the piping with a sand and paint mixture.

Other data have been analyzed in the course of this survey to determine the application of this relationship and sample plots are shown in Fig. 3. If the linear relationship of k and f may be assumed to pass through the origin, then



Symbol	Source	Approx. D - in.	ε/D Estim.	Approx. $N_{Re} \times 10^{-5}$
Tests with Air:				
x	Brown[29]	12	.0005	2
+	Busey[30]	12	.0005	3
•	Cooke and Statham[31]	8	.0008	1
	Madison and Parker[26]	3	.002	5
	"	6	.001	1
○	"	12	.0005	2
	"	33	.0002	5
□	O. E. Parker[32]	12	.0005	3
	Stuart etc.[13]	7	.0009	2
Δ	Weske[10]	5	.00004	2
	"	5	.00004	5
	"	5	.00004	8
▼	Wirt[4]	3	.002	8
Tests with Water:				
◇	Nippert[6]	1.4	.0002	2

FIG. 5. LOSS IN TOTAL EQUIVALENT HYDRAULIC DIAMETERS FOR 90-DEG ELBOWS OF SQUARE CROSS SECTION

Symbol	Source	Approx. D - in.	Estimated ε/D	Approx. $N_{Re} \times 10^{-5}$
Tests with Air:				
x	Frey[11]	7.9	.0008	2.5
	Hellman & MacArthur[33]	6.7	.001	1.4
○	Madison & Parker[26]	4.8-5.7	.001	0.8
◇	McLellan & Bartlett[34]	3.8	.002	3
Δ	Weske[10]	4.0-4.6	.00004	2
▲	"	4.0-4.6	.00004	5
	Willard[28]	9.0	.0007	2.6
▼	Wirt[4]	2.5-4.7	.002-.001	8
Tests with Water:				
	Bombach[12]	5.3	.002	1.5
□	Nippert[6]	0.6-1.4	.005-.002	2
—	Indicates higher of 2 values of R/W shown in each plot			

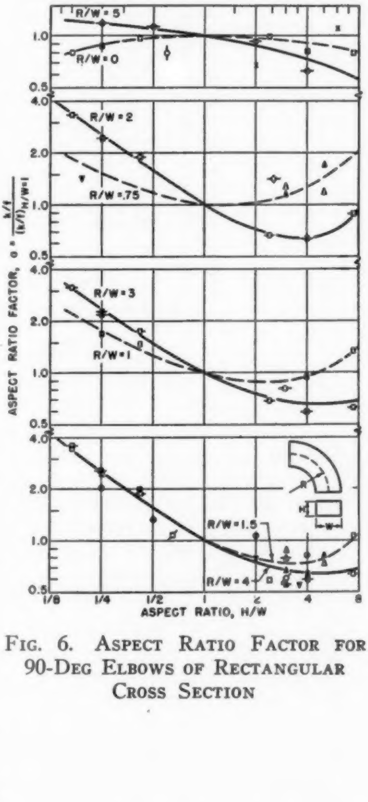


FIG. 6. ASPECT RATIO FACTOR FOR 90-DEG ELBOWS OF RECTANGULAR CROSS SECTION

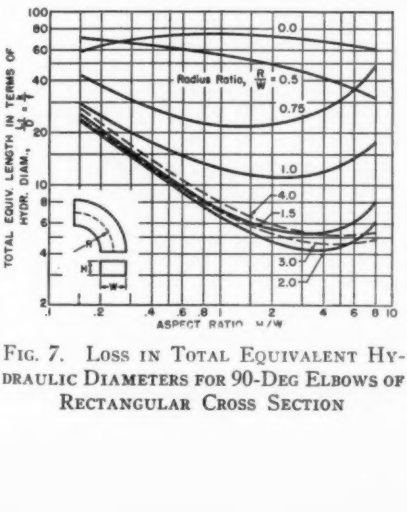


FIG. 7. LOSS IN TOTAL EQUIVALENT HYDRAULIC DIAMETERS FOR 90-DEG ELBOWS OF RECTANGULAR CROSS SECTION

the ratio k/f is a constant for a given elbow. Thus an opportnuity is offered for further reducing the number of variables for an empirical correlation of data.

Each curve of Fig. 3 is for a different elbow; the various values of the friction factor for each were obtained by changing one or more of the variables ϵ, D, or V as indicated. Size variation is shown by the data of Madison and Parker[26], velocity variation by the data of Stuart, Warner and Roberts[27], and combined effects of variable roughness and velocity by the data of Kirchbach[23] and Schubart[24]. Data of Willard[28] for compound elbows include combined

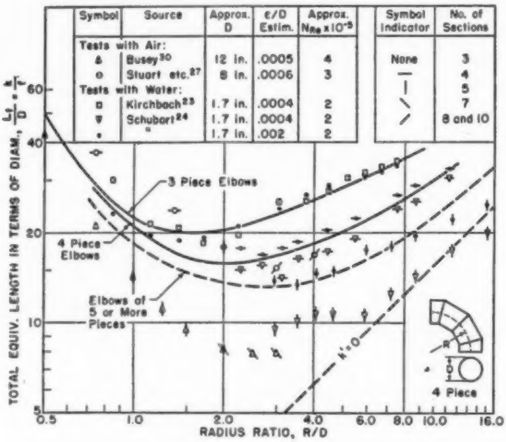


FIG. 8. LOSS IN TOTAL EQUIVALENT DIAMETERS FOR 90-DEG ELBOWS OF ROUND CROSS SECTION

effects of variable size, roughness, and velocity; these tests were made at velocities from 1000 to 7000 fpm on sheet metal and concrete models of elbows at the base of the Holland Tunnel air supply ducts. An illustration of the use of k/f is seen in Fig. 4 where the results for the models tested by Willard[28] are brought together by plotting k/f against Reynolds number.

Equivalent Length: Other data plotted in the manner of Fig. 3 indicate that k/f is constant within about ± 15 percent for the range of friction factors represented in the investigations surveyed. (This will be discussed further in the Appendix*.) The demonstration, that k/f may be considered essentially constant, justifies the assumption frequently made in design calculations that elbow resistance may be expressed in terms of *equivalent length* of straight duct.

The loss due to a fitting may be expressed as $h = k(V^2/2g)$ and the friction loss of a length of duct L_t which imposes a resistance to flow equivalent to that

* Material not essential to the presentation of the correlation will be found in the Appendix, including discussion regarding testing and presentation methods, and comparisons with previous surveys.

of the fitting is $h_t = f(L_t\ V^2/D\ 2g)$. Equating h and h_t yields

$$k/f = L_t/D \qquad (5)$$

or the ***total equivalent length in terms of hydraulic diameter.***

Effect of Radius Ratio: If the concept of equivalent hydraulic diameters is adopted, the fourteen variables are reduced to three dimensionless groups which must be dealt with in the empirical correlation of data:

$$k/f \text{ is a function of } (R/W,\ H/W) \qquad (6)$$

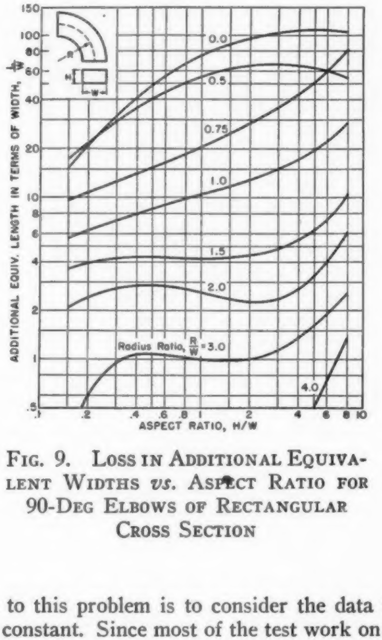


FIG. 9. LOSS IN ADDITIONAL EQUIVALENT WIDTHS *vs.* ASPECT RATIO FOR 90-DEG ELBOWS OF RECTANGULAR CROSS SECTION

The usual approach to this problem is to consider the data for the case where one of the groups is constant. Since most of the test work on rectangular elbows has been for those of square cross section, the effect of radius ratio was first investigated at an aspect ratio of unity. These data are plotted in Fig. 5 and include test values from various investigators (References 4, 6, 10, 13, 26, 29, 30, 31, 32).

A zone containing most of the data was established and a mean curve arbitrarily drawn. The zone of Fig. 5 covers values of k/f within ± 20 percent of the mean curve. The line for $k' = 0$ indicates the magnitude of the loss for a straight length of duct l' equal to the curved centerline of the elbow. Aside from errors in experimental technique, the scatter of data may be due to several reasons: differences in approach or discharge conditions, differences in the construction of elbows, inaccuracies in the determination or estimation of the friction factor, and limitations of the equivalent length concept. (See Appendix.)

Effect of Aspect Ratio: Preliminary plots of k/f against aspect ratio for constant values of radius ratio indicated that the scatter of data for different investigators was so great that trends could not be established. Instead, the data of each investigator (References 4, 6, 10, 11, 12, 26, 28, 33, 34) were plotted relative to the loss obtained for an elbow of square cross section. Thus the *aspect ratio factor a* is the ratio of k/f at a given aspect ratio to k/f at an aspect ratio of unity.

$$a = \frac{k/f}{(k/f)_{(H/W=1)}} \quad \ldots \ldots \ldots \ldots \ldots \quad (7)$$

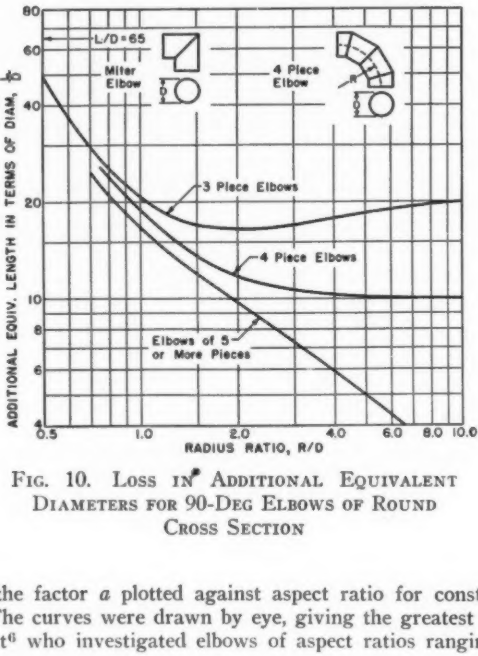


FIG. 10. LOSS IN ADDITIONAL EQUIVALENT DIAMETERS FOR 90-DEG ELBOWS OF ROUND CROSS SECTION

Fig. 6 shows the factor a plotted against aspect ratio for constant values of radius ratio. The curves were drawn by eye, giving the greatest weight to the data of Nippert[6] who investigated elbows of aspect ratios ranging from 1/7.5 to 7.5.

Having established the trend of the aspect ratio factor, these values were combined with k/f from Fig. 5 to give the summary plot of Fig. 7. A consistent family of curves is thus obtained for elbows having a finite inside radius. The form of the curve is apparently altered for 0.5 radius ratio by the presence of the sharp inside corner; the elbow at zero radius ratio is a miter elbow, having sharp corners at both the inside and outside.

Elbows of Round Cross Section: Many investigations have been made on losses in elbows of round cross section, covering radius ratios higher than those tested with rectangular section. Data are shown in the Appendix for losses in pipe bends, as distinguished from elbows built up of straight sections. Geome-

tries of built-up elbows have been expressed in terms of radius ratio and their loss data plotted in Fig. 8. The curves for three- and four-piece elbows were drawn by eye as the best line through the data. Scatter of data for elbows of five or more sections was such that the curve for pipe bends (established in the Appendix) was reproduced as representative of elbows of more than four pieces.

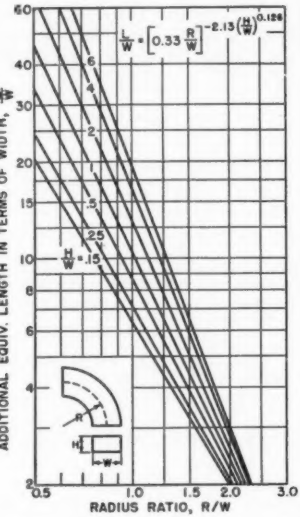


FIG. 11. LOSS IN ADDITIONAL EQUIVALENT WIDTH *vs.* RADIUS RATIO FOR 90-DEG ELBOWS OF RECTANGULAR CROSS SECTION

PRACTICAL INTERPRETATION OF DATA

Additional Equivalent Length in Terms of Width: In the calculation of elbow losses for duct design, use of total equivalent length in terms of hydraulic diameters L_t/D necessitates the determination of the hydraulic diameter D for each elbow. Thus it is believed more convenient for the design of rectangular duct systems to have the total equivalent length of duct expressed in terms of width W. The number of total *equivalent widths* is then

$$\frac{L_t}{W} = \frac{L_t}{D} \cdot \frac{D}{W} = \frac{L_t}{D}\left(\frac{2}{1 + W/H}\right) \quad \text{(8)}$$

by the definition of hydraulic diameter in terms of aspect ratio.

It is convenient in laying out duct systems to measure straight runs of duct to the intersection of their centerlines. When measurements are made in this way the *additional equivalent length* of duct necessary to represent the energy

loss is less than the *total* equivalent length by the length m-o-n of Fig. 2. Thus the ***additional equivalent length*** in terms of widths is

$$\frac{L}{W} = \frac{L_t}{W} - 2\frac{R}{W} \quad \ldots\ldots\ldots\ldots\ldots \quad (9)$$

Through Equations 8 and 9, the curves of Figs. 7 and 8 have been converted to additional equivalent length and are replotted in Figs. 9 and 10, respectively. The loss for a miter elbow is also shown in Fig. 10; data[6, 24, 27, 35] used in establishing this value are consistent within ±10 percent.

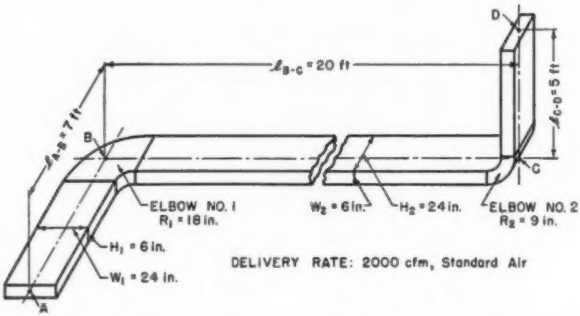


FIG. 12. PORTION OF DUCT SYSTEM FOR SAMPLE PROBLEM

Fig. 11 is offered for use where odd radius ratios are encountered. These curves correspond to the exponential relation,

$$\frac{L}{W} = \left[0.33\,\frac{R}{W}\right]^{-2.13\left(\frac{H}{W}\right)^{0.126}} \quad \ldots\ldots\ldots\ldots \quad (10)$$

and approximate a cross-plot of Fig. 9 within about ±20 percent.

Illustrative Problem: Use of Fig. 11 is illustrated by the following problem.

Given the portion of a duct system shown in Fig. 12, it is required to determine the pressure loss between points A and D. Air at standard conditions is being supplied at the rate of 2000 cfm in a 6 by 24-in. galvanized duct of average construction. Elbows, Nos. 1 and 2 have centerline radii of 18 and 9 in. respectively.

For elbow No. 1 the radius ratio is $\frac{R_1}{W_1} = \frac{18}{24} = 0.75$ and the aspect ratio is $\frac{H_1}{W_1} = \frac{6}{24} = 0.25$. The additional equivalent length for elbow No. 1 in terms of W is obtained from Fig. 11: $(L/W)_1 = 11.5$. Thus $L_1 = 11.5 \times 24/12 = 23$ additional equivalent feet. Similarly for elbow No. 2, the radius ratio is $\frac{R_2}{W_2} = \frac{9}{6} = 1.5$ and the aspect ratio is $\frac{H_2}{W_2} = \frac{24}{6} = 4.0$; Fig. 11 gives $(L/W)_2 = 6$, so $L_2 = 6 \times 6/12 = 3$ additional equivalent feet.

The total length of the straight runs from A to D is $l = l_{A-B} + l_{B-C} + l_{C-D} = 7 + 20 + 5 = 32$ ft and the additional equivalent length due to the elbows is $L = L_1 + L_2 = 23 + 3 = 26$ ft. Thus the equivalent length of the system from A to D is $l + L = 32 + 26 = 58$ ft of 6 by 24-in. duct.

The diameter of a circular duct, equivalent in friction and capacity to this rectangular duct, is 12.4 in. as given by the table of circular equivalents in The Guide[36]. At a delivery rate of 2000 cfm, the A.S.H.V.E. Friction Chart[36] gives a loss of 0.6 in. of water per 100 ft of 12.4 in. diameter duct. Thus the loss from A to D is $0.6 \times 58/100 = 0.348$ in. of water.

Comments: An exact correlation of existing data is practically impossible owing to differences in test methods, flow conditions, and elbow construction. No single investigation has covered the entire field of flow conditions and elbow geometry and construction. Each has instead, been confined to a relatively limited range of application. The effects of Reynolds number and upstream velocity distribution on the energy loss in elbows have not been accurately established in this survey because of the inadequacy and inconsistency of existing data.

It is believed that the curves presented in this paper for energy losses in 90-deg duct elbows are as accurate as may be deduced from the data presently available. If data of greater accuracy are desired, a comprehensive experimental program would seem necessary.

The survey should be extended to cover losses in vaned elbows, compound elbows, and elbows involving change of area.

Acknowledgment

The author acknowledges with thanks the help of all those who cooperated in assembling information and the suggestions given by his colleagues of the A.S.H.V.E. Research Laboratory staff.

APPENDIX

Test Methods

The experimental methods which have been employed in the determination of losses in elbows may be divided into two basic types, which will be termed the *deduction* method and the *interposition* method.

In the *deduction* method the loss was measured across a duct system containing an elbow between two straight runs. The loss due to the elbow was then obtained by deducting from the total the *calculated* friction over the straight length.

In the *interposition* method the energy loss between two stations on a straight run of duct was obtained at a given velocity head; then an elbow was interposed in the straight run and the loss obtained between the same two stations for the same velocity head. The difference of these two losses was taken as the loss due to the presence of the elbow. In some investigations the second station was the atmosphere.

Results of different investigations cannot be considered strictly comparable, unless the conditions at the measurement stations were those characteristic of fully developed flow. The following discussion will consider these conditions (*a*) upstream of the bend, (*b*) downstream of the bend, and (*c*) in the measurement of pressure gradients.

It has been shown by Yarnell and Nagler[37] that the upstream velocity distribution plays an important part in affecting the loss due to an elbow. Approach conditions were no doubt quite different for certain of the investigations surveyed, notably those of Busey[30] and Wirt[4]. The tests by Busey[30] were made on elbows which were only three diameters downstream of a plenum chamber, and velocity measurements were taken with a pitot tube located about 2½ diameters from the chamber; air discharged to the atmosphere three diameters downstream from the elbow. Here the disturbance due to duct entry from the plenum chamber possibly produced a dis-

turbance which persisted to the elbow. The velocity profile entering the elbow was flat in the tests of Wirt[4], where the elbow was immediately preceded by a nozzle-like reduction.

In several investigations where the *interposition* method was used with the atmosphere as reference, a very short length of duct (three or four diameters) was used downstream; these investigations included those of Parker[32], Busey[30], Madison and Parker[26], Wirt[4], and Nippert[6]. The possibility exists in these cases that the discharge to the atmosphere so close to the elbow may have had an appreciable effect upon the measured loss. It is interesting to observe in Fig. 5 that in general the k/f value for tests run in this manner are lower than for the other investigations. The work of Schubart[24] and Beij[38] point to the fact that losses due to the elbow persist for as many as 50 diameters downstream in round pipes, particularly following bends of large radius ratio where the secondary flow is not broken up by large

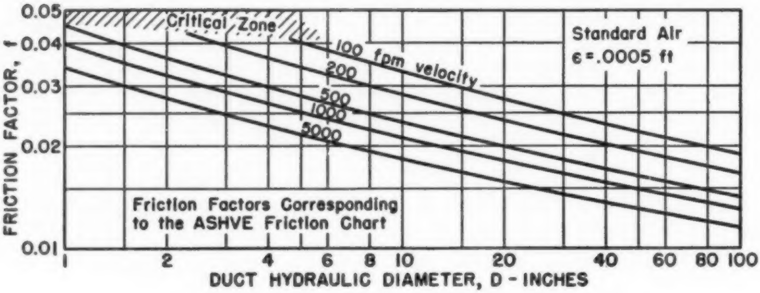


FIG. A. FRICTION FACTORS FOR SHEET METAL AIR DUCTS

velocity discontinuities. With the exception of the work of Nippert[6] the data in Fig. 5 for the investigations mentioned previously, appear to be progressively lower in the zone approaching high radius ratios.

A variation of the *deduction* method uses the difference in the height of the extrapolated pressure gradients upstream and downstream; this was the method used by Stuart, Warner and Roberts in their work[13, 27]. In order to yield the true loss due to the bend, the pressure gradients for incompressible flow must be parallel and must be measured far enough upstream and downstream to avoid influence of the elbow on the respective slopes.

There are several sources of direct error which may have been present to some degree in the experimental investigations surveyed; poor location of pressure measuring stations with respect to local disturbances, inaccurate pressure measurement, inaccurate fluid metering, and fluid leakage to or from the duct system.

REDUCTION OF DATA

Data reported in the literature were reduced when necessary to k and k/f; the net loss coefficient k' differing from k only by the friction calculated over the axial length of the elbow.

$$k = k' + f\,l'/D \quad \text{(A-1)}$$

Where test values of friction factors were not indicated, f was obtained by the chart of Moody[39], with the roughness ϵ estimated according to the recommendations of

Madison and Elliot[40] and Moody[39]. Friction factors for galvanized duct of average construction are shown in Fig. A corresponding to the roughness selected by Wright[41] for the construction of the A.S.H.V.E. Friction Chart[36].

Wirt[4] employed a slightly different method of calculating the loss coefficient, making a recalculation necessary for comparative purposes in this survey. This recalculation is described.

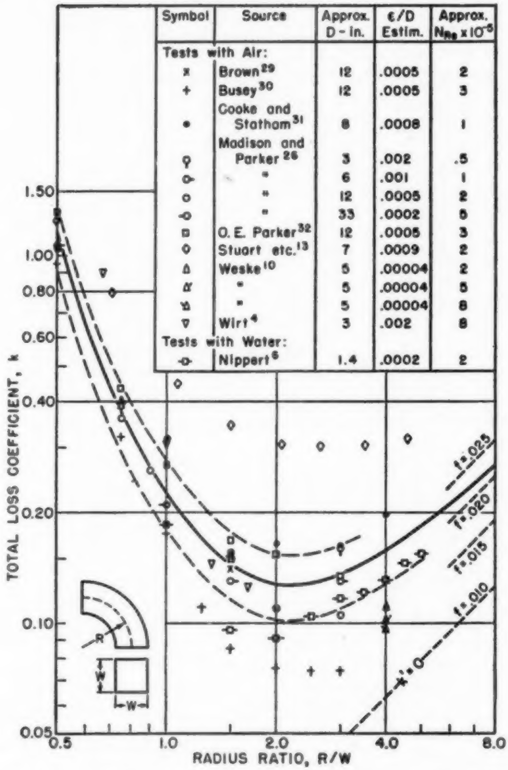


FIG. B. TOTAL LOSS COEFFICIENTS FOR 90-DEG ELBOWS OF SQUARE CROSS-SECTION

The experimental technique used was the *interposition* method with the atmosphere as reference. The total pressure was measured just upstream of a nozzle to which the elbow was mounted. Symbols used in this discussion are defined as follows:

p = static pressure at exit of nozzle.
q = dynamic pressure at exit of nozzle.
p_0 = total pressure measured upstream of nozzle.

Symbol	Source	Approx. D - in.	ε/D Estim.	Approx. $N_{Re} \times 10^{-5}$
Tests with Air:				
●	Cottignies[35]	5.9	.001	4
▫	Weske[10]	6	.00004	2
-o-	"	6	.00004	6
✦	Wirt[4]	3	.002	8
Tests with Gasoline:				
+	Bridgeman[43]	0.5	.0015	.3
Tests with Water:				
▼	Beij[38]	4	.001	.3
◆	"	4	.001	3
x	Fosdick[44]	6	.0003	5
■	"	6	.0003	2
▫⁄	Freeman[45]	4	.0005	.5
▫\	"	4	.0005	5
▫/	"	8	.0005	1
▫	"	8	.0005	5
o⁄	Hofmann[21]	1.7	Smooth	1
o\	"	1.7	Smooth	2
o/	"	1.7	.002	1
o	"	1.7	.002	2
▲	Nippert[6]	1.3	.0002	2
●	Schoder[46]	6	.002	1
●	"	6	.002	4
ϕ	Yarnell[47]	6	.0001	4

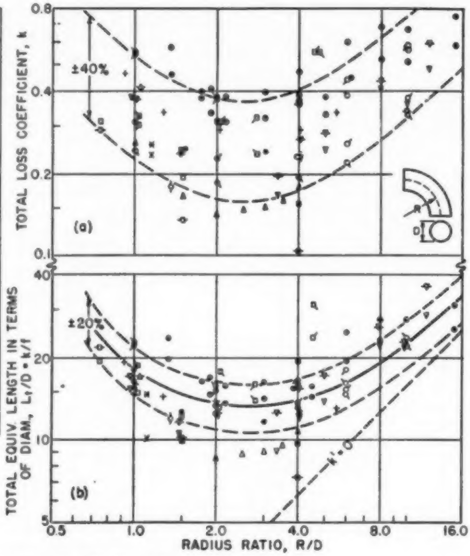


FIG. C. LOSS FOR 90-DEG PIPE BENDS

(a) In Terms of k (b) In Terms of k/f

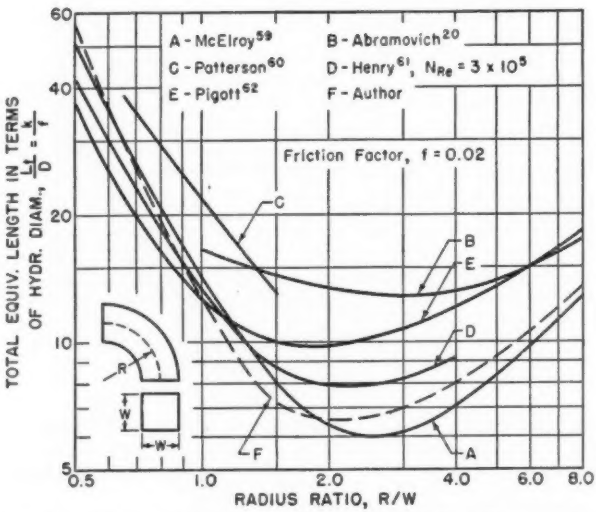


FIG. D. COMPARISON OF SURVEYS OF LOSSES IN 90-DEG ELBOWS OF SQUARE CROSS-SECTION

p_a = atmospheric pressure.
w = mass rate of flow.
R = gas constant.
n = exponent in equation: p/ρ^n = constant (n = 1.4 for air at normal temperature and pressure).
T_0 = absolute temperature (T_0 = 460 + 180 F, average air temperature for tests).
subscripts: $_1$ condition for nozzle discharging to atmosphere.
$_2$ condition for four-diameters duct following nozzle.
$_3$ condition for elbow and duct following nozzle.

The coefficient of loss for the elbow and the short duct was given by Wirt as:

$$k_W = \frac{p_{o3} - p_{o1}}{p_{o1}} \quad \text{(A-2)}$$

However, taking account of compressibility effects, which enter at the very high velocities used by Wirt, the expression for the total coefficient of loss for the elbow alone is

$$k = \frac{p_{o3} - p_a}{q_3} - \frac{p_{o2} - p_a}{q_2} \quad \text{(A-3)}$$

By introducing the definition, $q = p_o p$, Equation (A-3) becomes in terms of pressure ratios

$$k = \left(\frac{1 - p_a/p_{o3}}{1 - p_3/p_{o3}}\right) - \left(\frac{1 - p_a/p_{o2}}{1 - p_2/p_{o2}}\right) \quad \text{(A-4)}$$

For the determination of the pressure ratios, p/p_o, isentropic flow was assumed and thus the expression

$$\frac{w}{Ap_o} = \left(\frac{p}{p_o}\right)^{1/n} \sqrt{\left(\frac{2g}{RT_o}\right)\left(\frac{n}{n-1}\right)\left[1 - \left(\frac{p}{p_o}\right)^{\frac{n-1}{n}}\right]} \quad \text{(A-5)}$$

Since the paper by Wirt[4] gives the relation between w and p_o for a 3 x 3-in. elbow at conditions 1, 2, and 3, it was possible to deduce the left side of Equation (A-5). The corresponding values of p/p_o were obtained from a tabulation[42] of the pressure ratio functions. Then, by means of Equation (A-4), k was determined.

For the built-up elbows (Fig. 8) tested at the *Munich Hydraulic Institute*[23, 24] radius ratios were derived from the centerline length S of each segment and the angle α of each miter deflection by the following equation:

$$\frac{R}{D} = \left(\frac{1 + 1/\cos\alpha}{2 \tan \alpha}\right)\left(\frac{S}{D}\right) \quad \text{(A-6)}$$

METHODS OF EXPRESSING THE ENERGY LOSS

A discussion is in order of the limitations of the equivalent length concept when velocity and roughness are variables.

Data which were available over a range of friction factors, regardless of how the change was affected, were plotted as in Fig. 3. The relation could be considered substantially linear so that the equation of the straight line was

$$k = \left(\frac{dk}{df}\right) f + (\textit{constant}) \quad \text{(A-7)}$$

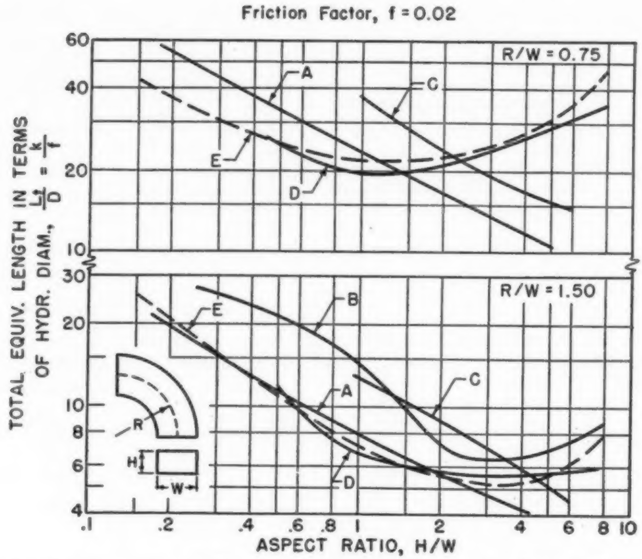


FIG. E. COMPARISON OF SURVEYS SHOWING EFFECT OF ASPECT RATIO ON LOSS IN 90-DEG ELBOWS OF RECTANGULAR CROSS-SECTION

A—McElroy[59] B—Abramovich[20] C—Patterson[60]
D—Henry[61], $N_{Re} = 3 \times 10$ E—Author

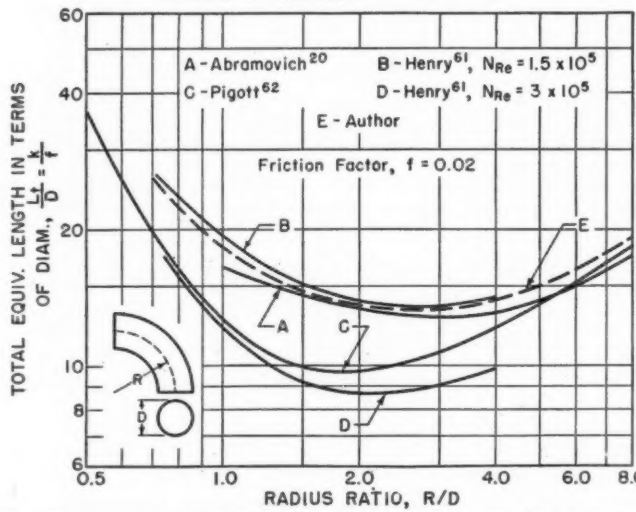


FIG. F. COMPARISON OF PREVIOUS SURVEYS OF LOSSES IN 90-DEG PIPE BENDS

If this *constant* is zero then k/f is constant at the value of the slope; however, many of the data showed that this constant was not zero. Although this analysis is quite sensitive to errors in determining the friction factor, several generalizations may be made. When the change in friction factor was due largely to change in roughness the constant appeared to be negative, and when the change of velocity produced the variation in friction factor the constant was generally positive. In cases where backflow was appreciable k was indicated to be nearly constant. The slope dk/df was even negative for certain tests[10] as low radius ratio and high aspect ratio.

The range of friction factors encountered in practice is quite limited so that the variation of k/f for a given geometry is not great. With few exceptions[24, 28, 45], k/f was found to be constant within ± 15 percent for the range of friction factors represented by tests.

An analysis to determine the effect of velocity upon k was made with the data of several investigations[10, 26, 32] of rectangular elbows. The coefficient k was plotted *vs.* the Reynolds number on logarithmic coordinates and the exponent x determined for the relations:

$$k \text{ is proportional to } N_{Re}^{x}$$
$$\text{and } \Delta p \text{ is proportional to } V^{2+x} \quad \text{(A-8)}$$

In general x was found to be positive for low radius ratios and high aspect ratios where backflow is expected. It became negative for higher radius ratios. The data of Weske[10] indicate a sharp rise in loss at high Reynolds numbers for aspect ratios near five, particularly at low radius ratios; in the absence of other supporting data mean curves were drawn in Fig. 6.

Loss coefficients for elbows of square cross-section are plotted in Fig. B. At low radius ratio, these data in terms of k are somewhat more consistent than when plotted in terms of k/f (Fig. 5). The variation of the exponent x above tends to indicate that this would have been more noticeable had a greater range of Reynolds numbers been investigated. The effect of size, however, is shown by the spread of Madison's and Parker's data[26] at radius ratios of 1.5 and 3.0.

Loss data are plotted in Fig. C for pipe bends having no appreciable discontinuities of the inside surface. Flanged elbows and tubing bends are included; screwed elbows[45, 48, 49, 50, 51] are not included. The range of roughness represented in the data of Fig. C is greater than that for the rectangular elbows of Figs. 5 and B, to the extent that a zone of ± 40 percent is required to represent these data in Fig. C(a). Use of the equivalent length concept brings these data substantially within a zone of ± 20 percent as indicated in Fig. C(b). Certain other tests[52, 53, 54, 55, 56, 57, 58] for pipe bends have not been included in these plots.

Regarding the different methods of presenting data, it may be said that the most reliable and fundamental method is the plotting of Δp *vs* velocity for each elbow geometry and roughness. This would require many tests and many design curves to cover the types of elbows used. However, use of the curves proposed in this paper will facilitate loss calculatons without seriously sacrificing accuracy. When more comprehensive data become available over a wide range of conditions, a Reynolds number characteristic will probably be established for the loss in each geometry of bend, particularly at low radius ratios and high aspect ratios.

COMPARISON WITH PREVIOUS SURVEYS

It is interesting to compare the summaries of this survey with those obtained in other *analytical surveys*. The curves of Figs. D, E and F have been prepared to show this comparison. Summaries reported in terms of k and k' were plotted by assigning the friction factor $f = 0.02$; other values of f serve only to displace these curves vertically.

McElroy[59] has proposed the following exponential equation for the net loss coefficient of rectangular elbows, with particular application to bends in mine airways:

$$k' = 0.25\,(R/W)^{-2}\,(H/W)^{-1/2} \quad \text{. . . . .} \quad \text{(A-9)}$$

There is fair agreement between this relation and the author's summary for aspect ratios of unity and below.

The summaries of Abramovich[20], Patterson[60], and Henry[61] were in the form of curves; those of Abramovich were to apply to both rectangular and round cross sections.

In a recent analysis, Pigott[62] has recognized the importance of the roughness and Reynolds number in influencing elbow losses and has empirically deduced the following expression for the net loss coefficient in round and square elbows:

$$k' = 0.106\,(R/D)^{-2.5} + 2000\,f^{2.5} \quad \text{. . . . .} \quad \text{(A-10)}$$

The first term represents the net loss coefficient for a fictitious *dead smooth* elbow. The second term gives the increase due to the combined effects of roughness and Reynolds number; here also the friction factor was used to characterize these two factors.

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DISCUSSION

C. M. ASHLEY, Syracuse, N. Y. (WRITTEN): The author is to be commended upon the thoroughness with which the correlation of the available information on this subject has been made and the general excellence of the presentation. Of particular interest is the demonstration of the relationship between the friction factors and the energy loss in velocity head.

It is unfortunate that the main body of the paper did not include a discussion of the effect of approach and downstream conditions on the elbow efficiency since it is important to emphasize the possible adverse effect of two elbows in close succession or of other disturbing elements upstream or downstream.

It is evident that there is still room for a definitive test program and it is believed that this analysis has laid suitable ground work for such a program.

F. C. HOOPER, Toronto: Would it not be possible to extend this work to cover turns of all angles rather than only 90 deg bends? The success of the dimensional approach suggests that this might be achieved through a sound yet relatively simple relationship.

R. D. MADISON, Buffalo: A survey of this type is very difficult because most of the original data are not available and reference must be made to abbreviated reports. The author has done an excellent job in comparing the available material and pointing out values that are reasonable to use. When this work is extended to cover elbows with splitters and non-uniform elbows with turning vanes, a further contribution to the designer of duct systems will have been made.

Much of the available material has been on the basis of loss in percent of velocity pressure; data in these terms are shown in Figs. B and C of the Appendix. As has been pointed out by the author, the idea of equivalent length or equivalent additional length serves very well for comparing elbows of the low loss type. When it comes to elbows having a loss of ¼ to 1 velocity head loss, the velocity head loss method is more exact since the effect of size becomes negligible. Also the effect of velocity is minimized on this comparative basis. However, it is to be hoped that low loss elbows will be used where possible, and the method outlined in this paper becomes a logical solution to presentation of elbow losses.

CYRIL TASKER, Cleveland: Originally, the laboratory was to build a test setup in order to investigate the energy losses in elbows. Instead of doing this, however, it was decided, to carefully and critically review all the available information and to include not only information that is normally available to heating and ventilating engineers, but material that was not usually catalogued.

We have made this analysis, and it has been presented to you as the best available information. For the time being, these data, I think, represent an advance over what is generally available and I think they are in a form than can be used by most practicing engineers. These data, I think, are sound, and we plan to complete the rest of this study for other types of elbows in the future.

This research is an essential prerequisite to any experimental program. We hope to be able to continue to make this type of study as our work proceeds. The results of this work are being incorporated in THE GUIDE.

AUTHOR'S CLOSURE: In response to Mr. Ashley, the information available on the effects of approach and downstream conditions is not wholly conclusive. There re-

mains need for a comprehensive experimental investigation if these effects are to be definitely established for the range of such conditions encountered in practice.

Replying to Mr. Madison, it was felt that the extension of the survey to losses in vaned elbows and special elbows should prove very interesting. It has been established by the data of several investigators that a substantial saving in loss can be realized by the installation of vanes; however, the problem of correlation of these data becomes more difficult due to the number of additional variables introduced by tests on vanes of different forms and spacing in different elbows.

The survey showed that the loss for a given elbow may be reasonably generalized by either the equivalent length concept or the percent of velocity head method. The equivalent length concept was chosen for this presentation since the author believed that it was more consistent over a wide range of elbow forms and that the concept was simpler in application to design.

As Mr. Hooper suggests, losses in elbows other than 90 deg may be handled by retaining the term l'/R in Equation 2 and subsequent dimensionless relations. In this presentation however, the usual assumption of loss being proportional to the angle of bend is believed within the accuracy of the correlation.

In Memoriam 1950

Name	Joined
Charles P. Atkinson, Cleburne, Tex.	1949
James Baker, Montreal, Que., Canada	1944
Curtis H. Bevington, Chicago, Ill.	1936
Warren C. Bevington, Indianapolis, Ind.	1942
Charles R. Bishop *(Life Member),* Bronxville, N. Y.	1901
Leslie L. Bysom, Bremerton, Wash.	1938
Louis A. Calcaterra, Grand Rapids, Mich.	1944
Willis H. Carrier *(Life Member* and *Presidential Member),* Syracuse, N. Y.	1913
Bert C. Davis *(Life Member),* Elmira, N. Y.	1904
Robert B. Dickson *(Life Member),* Kewanee, Ill.	1919
William H. Driscoll *(Life Member* and *Presidential Member),* Jersey City, N. J.	1904
Ralph R. Emerson, Boston, Mass.	1922
Daniel J. Fagin, St. Louis, Mo.	1933
A. M. Feldman *(Life Member),* New York, N. Y.	1903
Robert H. Feltwell, Washington, D. C.	1905
W. E. Goodram, Hamilton, Ont., Canada	1939
Edwin A. Gordon, III, Austin, Tex.	1949
Matthew H. Goss, Detroit, Mich.	1938
William F. Graham, Toronto, Ont., Canada	1946
Arthur L. Henze, W. Lafayette, Ind.	1945
E. Vernon Hill *(Life Member* and *Presidential Member),* Chicago, Ill.	1912
Nelson B. Hubbard, Detroit, Mich.	1937
George P. Jackson, Chicago, Ill.	1949
James J. Kelley, Pittsburgh, Pa.	1949

In Memoriam 1950

(continued)

Name	Joined
John L. Larson, Cicero, Ill.	1949
Joseph A. Martocello, Philadelphia, Pa.	1934
John J. McDonald, Boston, Mass.	1927
John G. Mench, Park Ridge, Ill.	1945
John M. Moriarty, Arcadia, Calif.	1937
William W. Murphy, Springfield, Mass.	1930
Hugh T. Porter, New York, N. Y.	1944
E. R. Roach, Richmond, Va.	1941
Stanley F. Robertson, Baltimore, Md.	1949
Edgar R. Robinson, Westmont, N. J.	1947
Arthur M. Rosenblatt, Charleston, W. Va.	1938
John J. Shanahan, Norfolk, Va.	1946
Harry Simpson, Toronto, Ont., Canada	1945
Walter Spoffoth, Seattle, Wash.	1930
Donald R. Steeves, Toronto, Ont., Canada	1944
R. H. Thomas *(Life Member),* Glendale, Ohio	1920
Frank N. Tucker, New York, N. Y.	1926
Kimball Vance, Salt Lake City, Utah	1949
Alfred G. Wahlberg, Duluth, Minn.	1950
Albert E. Watts, Montreal, Que., Canada	1937

Dr. Willis H. Carrier

1876-1950

The "Chief", as Dr. Willis H. Carrier was affectionately known to his associates and friends, was called to his eternal reward on October 7, 1950. Dr. Carrier died in Cornell Medical Center of New York Hospital after having spent two weeks there as a patient for a check of a heart ailment. He was chairman-emeritus of Carrier Corp., Syracuse, N. Y., at the time of his death.

He was born on November 26, 1876 in Angola, N. Y., and attended Cornell University, graduating with an M.E. degree in 1901. He received his D.E. from Lehigh University in 1935 and his D.Sc. from Alfred University in 1942.

Dr. Carrier formed the Carrier Engineering Corp., at Newark, N. J., with J. Irvine Lyle and others in 1915, becoming its president. The firm later became the Carrier Corp. He served as president until 1931, when he became chairman of the board.

Dr. Carrier's interest in air conditioning had its origin in the problems of a Brooklyn color processor in 1902, when he was employed as an engineer by the Buffalo Forge Co. As a result of his inquiries into this problem, Dr. Carrier developed in 1903 what was known as a spray-type air washer, and in the course of this work became interested in the general problem of humidification and dehumidification. In 1905 he devised a spray-type air conditioner capable of heating, cooling, humidifying air, the basis for those which are on the market today.

It was as a result of this development that he left the Buffalo Forge Co. where he had become chief engineer in 1906, to pursue the studies that led him to the organization of the Carrier Engineering Corp. in 1915. In 1907 he had patented what was known as *dewpoint control,* a method of regulating humidity by controlling the temperature of the spray-washer in a washer or conditioner of air.

His researches led Dr. Carrier to an exhaustive study of the psychrometric phenomena, including the numerous factors related to the dehumidification of air by the use of mechanical refrigeration. He completed this study in 1911 and presented to the *American Society of Mechanical Engineers* a paper, *Rational Psychrometric Formulae,* which was ranked as fundamental engineering doctrine and upon which was built the science of air conditioning. Thus, Dr. Carrier had earned for himself the title of *father of air conditioning.*

Further studies brought about in 1914 the publication of *Fan Engineering,* which he edited, containing many tables and much data of basic importance. In

1912 he had produced a self-contained air conditioning unit, a combination of motor-driven fan and pump to send air through a spray of controlled temperature, which was probably the first such unit designed for railroad use.

In the ensuing years, Dr. Carrier made numerous contributions to air conditioning. These included the ejector system, first developed in 1916 for the tobacco, food, textile and other industries, and many of the principles used in conditioning the air in theaters and other large public gathering places.

Honorary membership was conferred upon Dr. Carrier in 1944. He was very active in the Society serving as its president, first vice president and second vice president, in 1931, 1930 and 1929, respectively. He was a Presidential member and served on the Advisory Council from 1932-1950. Active in New York Chapter affairs, he served as the Chapter's president and vice president in 1921-22 and 1920-21, respectively. He served on many important committees of the Society among which were:

1917-19	To Cooperate with U.S. Navy Dept.
1919	To Confer with Weather Bureau
1921-22	Synthetic Air Chart (chairman)
1921-23	Sub-Committee on Ventilation
1923	Membership (chairman)
1923	Sub-Committee on Subjects
1923-25	Executive
1923-28	Sub-Committee on Air Washers and Humidifiers
1924-25	Membership (chairman in 1925)
1925	Standards of Ventilation (chairman)
1925	To Develop a Ventilation Safety Code under the Procedure of *The American Engineering Standards Committee*
1926-27	Publication
1927-28	Committee on Research
1927-28	TAC to Cooperate with Rochester School Board (vice chairman)
1927-29	TAC on Temperature, Humidity and Air Motion (chairman)
1927-29	TAC on Standards of Ventilation (chairman)
1927-29	Guide Publication
1928	TAC on Garage Ventilation
1928	Meetings Program
1929-30	Executive
1932	Executive
1932	TAC on Re-Study of Comfort Chart and Comfort Line
1933	Nomenclature
1934	Ventilation Standards
1937	TAC on Treatment of Air with Electricity
1938	TAC on Air Cleaning and Atmospheric Impurities (Mechanical Procedures)
1938	Engineering in Its Relation to Public Health
1938	Educational Publicity
1939-44	TAC on Psychrometry
1942	F. Paul Anderson Award
1945	F. Paul Anderson Award
1945	War Service
1946	Nominating Committee
1947	History Committee

Dr. Carrier was a member of the *A.S.M.E.*, *A.S.R.E.* (president in 1927), *A.A.A.S.* (elected Fellow in 1930), Sigma Xi and Tau Beta Pi (Cornell University), *Cornell Society of Engineers* (president in 1939), Technology Club, Engineers Club, Cornell Club, Onondaga Golf & Country Club, and the Century Club.

Dr. Carrier received the following awards: John Scott Medal in January 1932, which was awarded by the City of Philadelphia; the F. Paul Anderson Medal in March 1932, awarded in recognition of outstanding contributions in heating, ventilating and air conditioning by the A.S.H.V.E.; the *A.S.M.E.* Medal in 1934; and Franklin Institute's award—the first Frank P. Brown Medal in 1941.

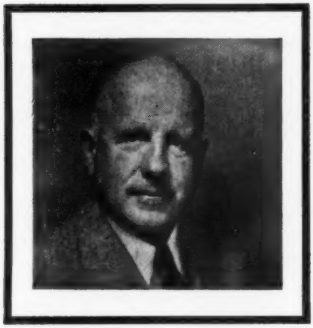
Dr. Carrier served as the United States' representative at the World Engineering Congress in Japan in 1929, and in 1940 was selected as a *modern pioneer of American invention* by the *National Association of Manufacturers.* In 1947 he was one of three Americans invited to address the centenary celebration of *Britain's Institution of Mechanical Engineers.*

Funeral services for Dr. Carrier were held at 11:00 a.m., Tuesday, October 10, at the Park Central Presbyterian Church, Syracuse, N. Y.

Dr. Carrier is survived by his wife, the former Elizabeth Marsh Wise, of 2570 Valley Drive, Nedrow, New York, and two sons, Earl G. (former member of Council) and Vernon G. Carrier.

William H. Driscoll

1879-1950



William H. Driscoll, past president and Life Member of the Society, passed away on September 6, in his 72nd year. Mr. Driscoll was vice president of the Carrier Corp., Syracuse, N. Y., and had recently been engaged in supervising the construction of a new plant for the company, having deferred his retirement for this project.

Mr. Driscoll was born in Jersey City, N. J., August 14, 1879, and received his early education in the public schools of that city. After graduating from high school he joined the A. A. Griffing Iron Co. of Jersey City as a clerk (1896-1897). He then spent two years with the Standard Steam Specialty Co. in clerical and sales work. In November 1899 he became a salesman for Frank C. McLain Co. and in February 1901 he joined John B. Clarke as a draftsman and later became superintendent of construction for John B. Clarke's Son, all in New York City.

Mr. Driscoll was determined to become an engineer and for a period of 12 years he attended night classes at Cooper Union, Columbia University and Pratt Institute. Previously he had attended New York Trade School and Mechanics Institute.

In 1910 he joined the Thompson-Starrett Co., Inc., and remained there for 27 years and when he left the company, he was vice president and a member of the board of directors. During his association with the company he was in charge of the installation of mechanical equipment, including heating, ventilating, air conditioning, and electrical work. He was intimately connected with the work of skyscraper construction during the period of development of this type of building. Among those with which he had close connection in New York were, the new Waldorf Astoria Hotel, Bank of Manhattan, Equitable and Woolworth Buildings, and many of the skyscrapers in Philadelphia, Pittsburgh, Cincinnati, Detroit, Chicago, St. Louis, and other cities.

Mr. Driscoll joined the Carrier Corp., Syracuse, N. Y., in 1937 as vice president in charge of construction and was with the company until the time of his death.

He joined the Society in 1904 and during the more than 45 years of his membership he was personally known to thousands of its members. He had the rare

privilege of association with the Society's founders, who were the pioneers in the practice of heating, ventilating, and air conditioning, as well as intimate contacts with men who are carrying on work in the field today. He was a man of great enthusiasm and energy and he brought to the Society his great organizing talent, as well as a great interest in its advancement. He played a leading role in many of the Society's most notable undertakings, such as the establishment of the Research Laboratory, the founding of the JOURNAL and THE GUIDE, and assisted in establishing a sound financial basis for Society operations.

Mr. Driscoll had a remarkable record of service to the Society and actively participated in its work for several decades. He served as president of the Society in 1926 and prior to that time had been first and second vice president and treasurer, and was a member of Council from 1918-1927, and served as chairman of the Committee on Research in 1924 and 1925. Among the numerous committee assignments were the following:

1917-24	Code for Testing Low Pressure Heating Boilers
1917-20	To Confer with *National District Heating Association* (chairman in 1919-1920)
	To Cooperate with U.S. Navy Dept.
1920-22	To Confer with *A.S.M.E.* Boiler Code Committee on Heating Boilers
1920-24	Steam and Return Main Sizes
1920-25	Executive Committee of Council
1921	Membership
1922-23	Finance (chairman in 1922)
	Guide Publication (chairman in 1927)
1923	Sub-Committee of Committee on Research on Ventilation
1926-27	Committee on Research (chairman in 1924 and 1925)
1927	Finance
1928	Revision of Constitution and By-Laws (chairman)
1930	Code of Minimum Requirements for Heating and Ventilation of Buildings
1932-35	Special Committee on Ventilation Standards (chairman)
1932-34	Code for Testing and Rating Condensation and Vacuum Pumps
1933-42	TAC on Corrosion
1933	Special Committee on Nomenclature
1936-43	TAC on Heating and Cooling Requirements of Buildings
1940-41	To Study Method of Selecting Society Officers and Council
1943-44	Committee on War Service
1946	F. Paul Anderson Award
1947	Special Society History
1948	Honorary Chairman of Committee on Arrangements for 1948 Annual Meeting in New York
1950	Research Plan Committee

Mr. Driscoll will be remembered as one of the Society's immortals for his devoted and unselfish service to its ideals and its advancement. He was one of the Society's builders and his ideals and leadership were an inspiration to many of his fellow members who will cherish the memory of his work and his wisdom. His name is indelibly inscribed on the rolls of the Society. In addition to his work for the Society he served for many years on the Board of Directors of the *Heating, Piping and Air Conditioning Contractors National Association.* He was active in the Knights of Columbus, was a member of the Holy Name Society of St. Aedan's Roman Catholic Church, and was a Life Member of the Society of the Friendly Sons of St. Patrick in the City of New York.

Surviving Mr. Driscoll are his wife, the former Mary L. Heffernan; two daughters, Elizabeth (Mrs. John Sherry) and Mary (Mrs. C. W. Miller); three sisters, Mrs. Mary Donnelly, Miss Jane Driscoll, and Mrs. Gertrude Marum; and three brothers, David, Martin and Daniel Driscoll to whom the members, the Officers, and the Council extend their profound sorrow.

A Solemn High Mass was offered in St. Aedan's Church at 10:00 a.m. on Saturday, September 9, and interment was in Holy Name Cemetery in Jersey City.

Dr. E. Vernon Hill

1876-1950

Dr. E. Vernon Hill, a Life Member and Past President of the A.S.H.V.E., died on May 15 in the University of Illinois Research Hospital at the age of 72. Dr. Hill, a physician, scientist, engineer, and a recognized authority in the science of aerology, was born on October 25, 1876, in Jamestown, Pa., and received his high school education at Andover High School, Andover, Ohio. Following his graduation, he became construction engineer for the New York Central Railway, which position he held until 1899. In 1899 he matriculated in chemistry at the University of Illinois, later attending Rush Medical College, Chicago, from which he was graduated in 1903 and then began the practice of medicine.

Dr. Hill was appointed medical inspector of the Chicago Board of Health in 1904 and the following year was certified as a sanitary inspector for the department. In 1912 he was made chief of the ventilation division of the Chicago Department of Health, and in 1917 was appointed chief of the department's sanitary bureau. He resigned from the health department to enter private consulting practice in 1922 and became active in the heating, ventilating, and air conditioning field. Since that time, Dr. Hill devoted himself to aerology and was the editor and owner of the *Aerologist.* Later he became interested in instrumentation and devoted some time to making instruments for air measurement and analysis.

Dr. Hill bcame active in Society affairs in 1912 and he served as second vice president in 1918, first vice president in 1919 and president in 1920. He served on the Society's Council from 1915 to 1921 and as a member of numerous technical committees. He was president of the Illinois Chapter in 1917 and 1918 and on the Chapter's Board of Governors in 1919. His wide interests and the extent

of his services to the Society are indicated by the record of membership on the committees on which he served. Among these were the following:

1915-20 Special Committee to Recommend a Standard Method for Testing Air Washers (chairman in 1915)
Publication
Charters for New Chapters
Engineering Cooperation
To Cooperate with U.S. Navy Dept.
Finance

1921-25 Committee on Research
Sub-Committee on Subjects
Revision of Constitution (chairman 1924)
Sub-Committee on Dust (chairman in 1922)
Sub-Committee on Ventilation Requirements for Public Buildings (chairman from 1923-25)
Synthetic Air Chart
To Consider Report of New York State Commission on Ventilation
Ventilation Standards

1926-30 Committee on Research (1926)
TAC on Temperature, Humidity and Air Motion
Guide Publication
To Cooperate with Rochester School Board
Standards of Ventilation
Sub-Committee on Ventilation Requirements for Public Buildings (chairman 1926-28)
TAC on Atmospheric Dust and Smoke
TAC on Air Cleaning Devices
TAC on Air Conditions and Their Relation to Health
Code for Minimum Requirements for the Heating and Ventilation of Buildings

1931-35 TAC on Air Conditions and Their Relation to Living Comfort
TAC on Atmospheric Dust and Air Cleaning Devices (Including Dust and Smoke)
TAC on Re-Study of Comfort Chart and Comfort Line
TAC on Air Conditioning in Treatment of Diseases (chairman 1935)
TAC on Comfort Standards for Summer Cooling (1935)
Special Committee on Ventilation Standards

1936-39 TAC on Air Conditioning in the Treatment of Diseases (chairman 1936)
TAC on Psychrometry
TAC on Relation of Body Changes to Air Changes (chairman 1937)

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